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CHERYL ANNE NEWPORT

B.Sc. (HONS) London

Evaluation of the design, construction and operation of a gas fuelled, engine driven heat pump, and its possible role in a U.K. market.

Thesis submitted for the degree of Doctor of Philosophy in Energy Research at the Open University, December 1983.

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ABSTRACT

This thesis describes the results of several years work on the design, construction, testing and evaluation of a gas fuelled, engine driven heat pump and its possible role in a future U.K. market.

In 1977 a joint venture was embarked upon by the Open University Energy Research Group, Lucas Aerospace and Milton Keynes Development Corporation to design, manufacture and install a gas fuelled heat pump in a rented house, and to monitor its performance in real operating conditions. It was one of a number of projects in the field of heat pump research and development supported by the Department of Energy.

Due to a delay in receiving research funds however, and because of the size of the unit, it was impossible to install the system in a suitable property and so it underwent an intensive laboratory test programme simulating various load patterns and operating conditions. The heat pump, using air as its source of heat was driven by a 360 cc single cylinder marine engine converted to run on natural gas. The work was completed in 1980 and the heat pump was found to work well and justified the design assumptions made, after allowing for the poor performance of the engine used. At 6°C (ambient) an output of 14 kW was achieved with an overall efficiency or C.O.P. of 1.1 which compares favourably with a typical seasonal gas boiler efficiency of around 0.65-0.70.

As well as giving a full technical description of the heat pump system, plus an analysis of the various individual components, the thesis looks at the historical development of heat pumps generally and briefly considers the applications to which heat pumps can be put in domestic, commercial and industrial markets, and the possible economies this would bring. It concludes by looking at the future work needed in order to achieve these ends.

ACKNOWLEDGEMENT

The author would like to pay tribute to some individuals, without whom this thesis would never have seen the light of day!

Firstly to my husband for his confidence in me, his continued moral support and bullying! To my parents for always believing in me and never giving up hope! To my supervisor for his encouragement and comments. To Richard McBride for the black and white plates and to the secretaries in the Physics Department for struggling with my writing and with a new word processor!

My thanks must also go to the 'team', without whom the project itself would not have been possible. To Bob Critoph, Bill Gregory (Lucas), John Butler, Bob Everett and Jeremy Chatfield. The project was funded jointly by the Open University, Lucas Aerospace, and the Department of Energy, via the Energy Technology Support Unit.

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CHAPTER 1 INTRODUCTION

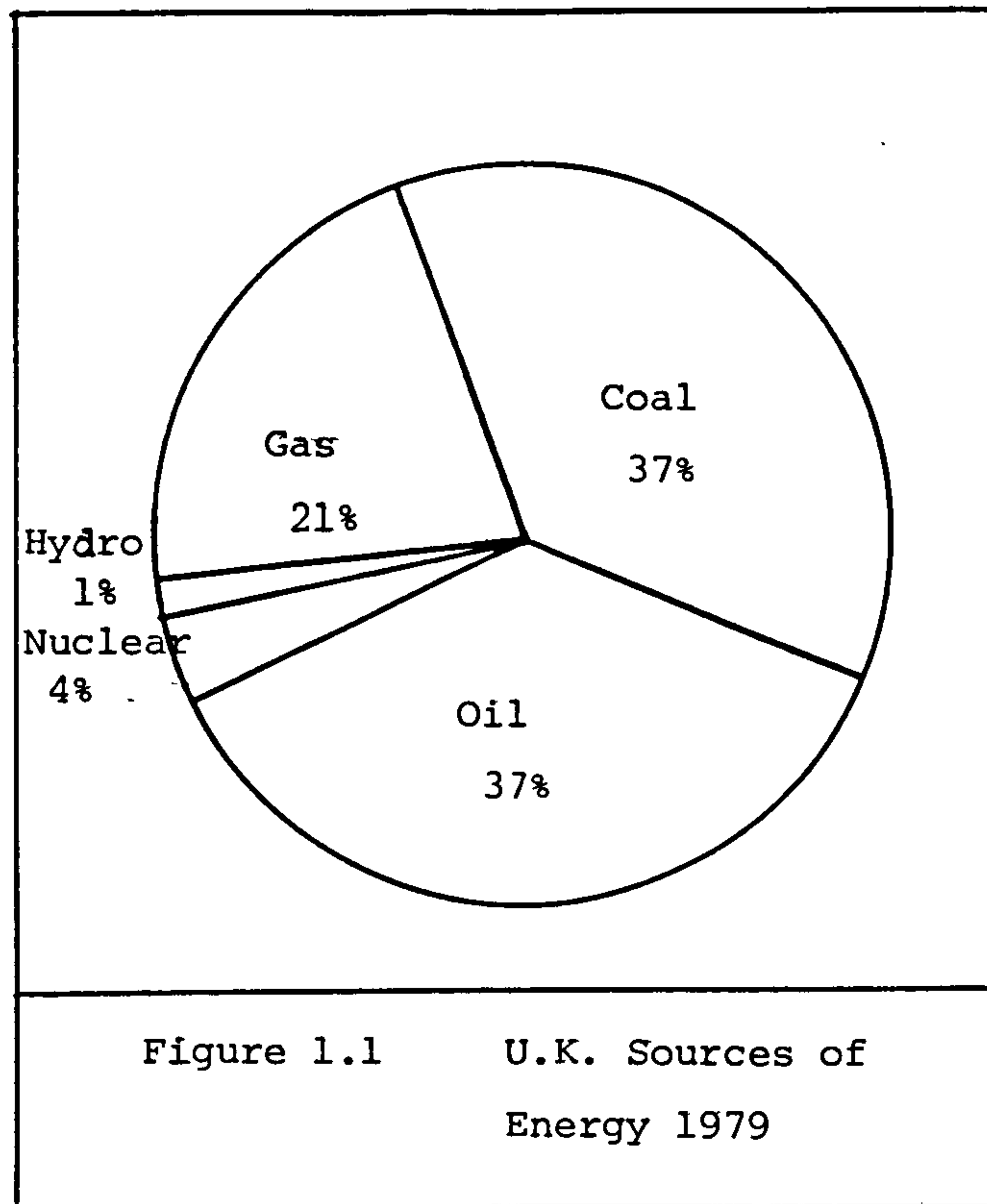
1.1 The Energy Scene

In our society today there is an ever increasing awareness of the vital role that energy - in all its various forms - plays in our lives. There is also the disturbing realisation that many of our traditional sources of energy have finite lifespans and, because of this, considerable time, money and effort are being devoted to finding new sources of energy and towards making better use of those we already have. [1,2].

This was not always the case. In the 1960's and early 70's energy policy was concerned almost entirely with questions of supply. The level of efficiency at which energy was used seemed of little or no importance since this was an era of cheap fuel in plentiful supply. In 1973, however, this rather rosy state of affairs was abruptly brought to an end by the Arab members of OPEC, with a dramatic warning to the rest of the industrialised world that the availability of their oil could no longer be taken for granted. By 1974 the price of oil increased fivefold, and since then it has become increasingly clear that energy prices are likely to continue to rise in real terms. Thus the scope and need for energy conservation, (although started in an aura of panic), is becoming of increasing importance.[3].

Most UK energy comes from its fossil fuels - coal and oil

with an increasing proportion of natural gas [4] as can be seen in Figure 1.1 .



North Sea oil and gas production puts the UK at present in the happy state of affairs of being virtually self sufficient in energy but this situation is not a cause for complacency. The oil and gas reserves of the North Sea are limited, and therefore the choice facing the British Government is either to export any surplus oil to improve the country's financial position, or to eke out the reserves by controlling the rate of depletion and so develop a breathing space in which to develop new or alternative sources of energy [5].

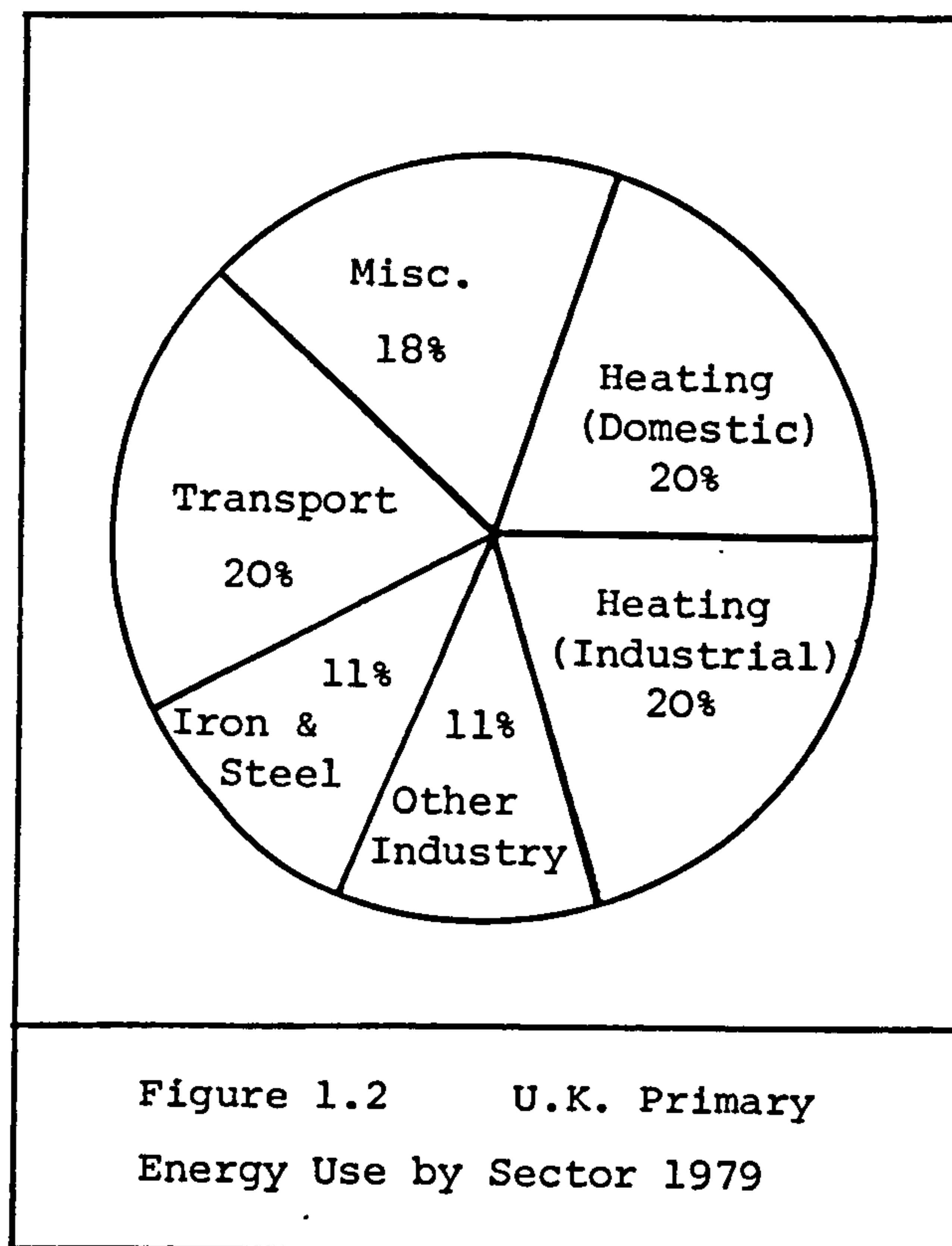
Considerable interest has been paid to the so called "alternatives" and they have been the subject of a number of government funded research programmes over the years. The wave energy programme began, in 1976, to examine four

different wave energy converters. Although energy from ocean waves has been identified as one of the major renewable resources available in the UK, the difficulties in harnessing that power to the grid has so far proved so enormous and so costly that the programme is progressing very slowly. In the Severn Estuary the UK has one of the most suitable tidal power sites in the world [6,7] but there are a number of environmental consequences to consider. Wind power [8] is less concentrated than wave power and more intermittent, giving rise to problems of storage and back-up supply. Nevertheless a 3 MW prototype aerogenerator is presently being built in the Orkneys by a Taylor Woodrow/British Aerospace consortium and the CEGB have a 400 kW experimental aerogenerator already in operation in West Wales. Solar Energy has been a major area of research over the past 10 years [9] and there have been numerous research projects dealing with a range of systems from simple flat plate collectors to focussing high temperature devices, photo-voltaic cells and storage systems. A great deal of work has also been carried out in the passive solar area and preliminary results indicate large savings for little extra building costs. [10]

To enable the UK to have the widest possible choice of alternative energy strategies continuing government support for the alternatives is essential, but irrespective of whichever strategy or combination of strategies is chosen, our supplies of fossil fuels can be maintained for as long as possible by the adoption of a widespread energy conservation

programme.

The use of primary energy by sector can be seen in Figure 1.2 and this shows clearly that a reduction in the energy requirement needed to heat a building, is the key to any successful conservation strategy, accounting for 40% of primary energy use in 1979.



Of the remainder, 22% of primary energy is used in the UK is for industrial processes and 20% is used for transport. Now in order to try and reduce these figures for energy consumption, there are two different strategies which can be adopted. The first is to reduce the demand for energy and thus reduce its end use by conservation measures (i.e. insulation, heat recovery etc.), and the second is to improve energy conversion and distribution efficiencies.

The UK has the poorest insulation standards in Europe and although building regulations have undergone a recent improvement these will only be beneficial to new constructions and will therefore, only have a small effect on the sector as a whole. Yet more revisions would still be needed to bring the regulations up to the European standards which themselves could be improved. Various grants have been made available to deal with existing private, public and industrial buildings to improve their thermal performance, but it is a very slow process [11,12]. In 1975 the British Government launched their "Save It" campaign and it was given wide media coverage to try and encourage the public's awareness of the need for energy conservation and of the existence of the various grants and schemes available to carry out conservation measures.

It soon became clear however, that a mere publicity campaign was not enough, for although the British public are conservative in many ways, they were apathetic to the need for energy conservation, with a widespread belief that an answer would be "found" to any energy shortage. Thus, in 1976, the Advisory Council on Energy Conservation set up its own Publicity and Education Working Group to consider how the UK public could be motivated to take further and more definite action to use energy more wisely and more efficiently. This was based on the belief that since everyone is an energy consumer in some way or other, then the aggregate of many individuals in all sectors adopting some energy saving actions, could cause a substantial reduction in

the national demand for energy.

The Working Group published their report in 1979 [13] and made many recommendations, several of which were immediately adopted and are still in existence today e.g. special training courses for energy managers in industry, the appointment of fuel efficiency officers in local government and the wide adoption of energy education from junior schools to University level. Even the Open University were given a £20,000 grant to produce a short course entitled "Energy in the Home" [14] which I helped to write. This was aimed at providing ordinary householders with sufficient information to understand both how they used energy in their homes, and how to determine which energy saving measures would be most cost effective for them given their own individual circumstances. Unfortunately there are still some of the Working Groups recommendations e.g. energy labelling of appliances which have not yet been adopted.

The second method of reducing energy consumption, as mentioned earlier, is to improve energy conversion and distribution efficiencies. The British Gas Corporation have contributed a great deal to improving the efficiency of existing appliances [15,16] and Government grants exist in the industrial sector to identify and replace inefficient boilers with better, more controllable systems. At present the net thermal efficiency of most modern coal or oil-fired power stations (i.e. electricity sent out compared to the fuel burned) is 33.7%, with nuclear stations ranging from

25%-38%. A typical medium sized combined heat and power (CHP) station will provide electricity at about 24% efficiency, and also produce water at a temperature of 70-90°C, suitable for space heating at nearly 50% efficiency. Thus the total efficiency is around 70% which explains why such schemes, used for district heating in large cities, are very popular in Europe and particularly so in Sweden and Denmark. A recent government working party in Britain [17] concluded that CHP schemes could undoubtedly save energy and that by converting a quarter of the UK's population to district heating would save about 20 mtce* a year, but are pessimistic that the economics of CHP are too high. Also included in this category of improved energy conversion and distribution is the development of new very efficient energy appliances. This is where my own work has concentrated, with the development of a gas engine driven heat pump.

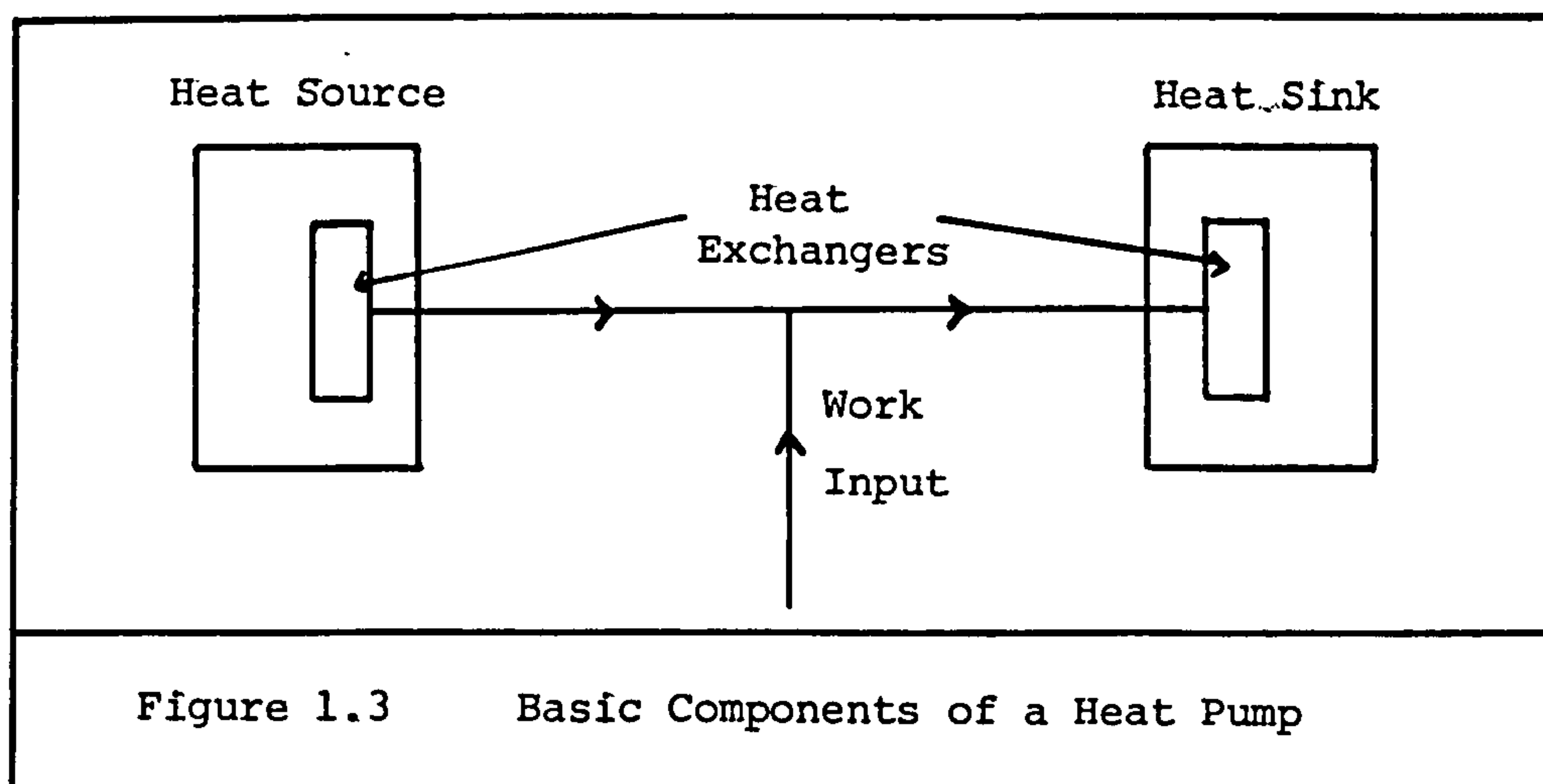
1.2 The Role of the Heat Pump

A heat pump is a device which extracts heat energy from a lower temperature source such as air, water, soil or rock and upgrades it, at the expense of some work, to a higher temperature. The principle is exactly the same as that used in the domestic refrigerator, except that here the higher temperature energy is useful and not wasted.

The essential components of any heat pump, shown in Figure 1.3, are the two heat exchangers, one to extract the heat

* Million tonnes of coal equivalent

from a low temperature source, and the other to eject the heat into a higher temperature heat sink. The heat is pumped from one to the other by several methods as I shall describe later, but all of them require some energy input to operate.



The efficiency of such a machine is known as its coefficient of performance (C.O.P.) and is the ratio of the output energy, Q to the work input energy W .

$$\text{C.O.P.} = \frac{Q}{W}$$

The heat pump is not a new device. It was first invented by Lord Kelvin in 1852, but due to the era of cheap and plentiful fuels which prevailed at the time it was not able to achieve a large foothold in the market. This situation changed however in 1973, and heat pumps experienced renewed interest. Indeed in 1975, a report published by the Building Research Establishment [18] estimated that widespread utilisation of heat pumps for heating purposes could reduce the national primary energy consumption by some 7%.

To date most research and development has concentrated on electric heat pumps. However these, whilst providing an

efficient use of electricity, do not necessarily provide the most efficient use of primary energy, due to the inefficiency of electricity generation and distribution in this country. For a given size of compressor the gas engine driven heat pump will always produce more heat than its electric equivalent since the engine's waste heat, in the form of hot exhaust gases, and water from the cooling jacket, can be recovered and used to supplement the heat pump's output.

The efficiency of the system in this case is termed the coefficient of fuel utilisation (cfu) which is then the ratio of total heat output divided by the gas input.

$$\text{cfu} = \frac{\text{total output}}{\text{gas input}}$$

Although the potential fuel saving by gas engine driven heat pumps is now well known, their application is still extremely slow and there are no companies as yet with a product in mass production.[†] A different story exists in Europe, in particular in West Germany, where there are now over 100 machines in use and more than 15 manufacturers. In comparison, manufacturers in this country are still just testing the water. However they are now receiving encouragement from British Gas who are trying to follow the example set by Ruhrgas, one of the West German supply companies.

In order to protect their future gas market Ruhrgas have funded research on gas engines and have set up a subsidiary

[†] Even now, in 1983, there are still only 20 prototypes in operation.

company in order to advise potential heat pump manufacturers and users. The car company Ford are also playing an active role in heat pump development by promoting the use of its 1.6 litre 2274E car engine for heat pump application [19]. This is backed by the Ford service network and they are also running training programmes and conferences for potential users, maintenance engineers etc.

The British Government too is playing its part, although perhaps not such a large role as European governments, under its Energy Conservation Demonstration Projects Scheme. This scheme provides grants for the application of new technology; new applications of existing technology and for monitoring the results. The work described later (Chapters 5, 6, 7) was funded under this scheme and, as such, formed part of a large body of work on gas engine driven heat pumps for the UK. [20]

The aim of the project was to design, construct and monitor the performance of a gas engine driven heat pump, in order to collect data and gain valuable experience of the difficulties associated with the operation of gas fired heat pumps. This was with a view to the design of a larger, commercial unit, which is believed to be a more economical unit at the present time. The objective was to verify the demonstration design assumptions by monitoring the heat pump under a wide range of operating conditions and to investigate the effect of different control strategies. Another valuable piece of information sought related to the maintenance and reliability

of engine driven systems, of which little is known, and also of noise levels which affect the future widespread acceptability of such heat pumps.

1.3 Outline of Thesis

A full description of the heat pump and the project is given in Chapters 5 and 6. The heat pump, using air as its source of heat, is driven by a 360cc single cylinder marine engine, converted to run on natural gas. The unit was designed at the Open University's Energy Research Group in collaboration with Lucas Aerospace Ltd. The unit was built at the Lucas Burnley plant and transferred to the Open University to undergo intensive performance testing.

The results of these tests are given in Chapter 7, but the heat pump was found to work well and justify the design assumptions made, having allowed for the poor performance of the particular engine. At 6°C (ambient), an output of 14kW was achieved with an overall efficiency (cfu) of 1.1, which compares favourably with a gas boiler's seasonal efficiency of around 0.65-0.70.

Chapter 2 looks more closely at the chequered history of the heat pump, as briefly mentioned earlier. The reasons are examined for its initial rise and subsequent demise, followed by a period of inactivity until its present upsurge of interest.

When designing a heat pump there are numerous choices to be

made - which heat pump cycle to use, and how to drive it? Which heat source and which refrigerant? The type of compressor and heat exchanger? All these various components are given careful consideration in Chapter 3 and the advantages and disadvantages of the various types noted.

Chapter 4 looks at the various applications to which heat pumps can be put in the domestic, industrial and commercial sectors and at the possible economies which such large a scale adoption of heat pumps could offer.

Chapter 8 concludes by considering the results obtained from the study carried out and recorded in Chapters 5, 6 and 7, and looking at the future work necessary to achieve the various applications discussed in Chapter 4.

Due to an unfortunate sequence of events including the break up of the research team, limitations and delay in funding of the project and difficulties in acquiring laboratory space, certain questions raised during the measurement and monitoring phase of the project had to be left unanswered, although the basic objectives of the project were achieved. Indeed, subsequent to the completion of the project, it was found that these identified discrepancies were not isolated incidents but were also being observed in other heat pump experiments. Unfortunately, by the time that this was realised, the equipment had been dismantled and further investigation was impossible.

CHAPTER 2 : REVIEW OF THE HEAT PUMP SCENE

2.1 Early History of the Heat Pump

The basic principle of the heat pump was first proposed by Carnot in 1824, deriving from his work on the Carnot cycle. Nearly thirty years later (1852) William Thomson (later to become Lord Kelvin) published a paper entitled "On the economy of the heating or cooling of buildings by means of currents of air". [1] This contained details of the first practical heat pump or "heat multiplier" as Thomson called it.

It was not until the 1920's that Thomson's ideas were expanded upon, firstly by Krauss [2] and then by Morley [3]. In 1930 Haldane [4] published details of the data he had collected from a selection of refrigeration plants and then calculated the actual COP's of these plants had they been used as heat pumps. These results, shown in Figure 2.1 overleaf, were extremely favourable, and as further proof, he constructed an experimental unit at his home in Scotland, using outside air as the heat source.

Following on from this work interest in the United States led to several privately financed, customized systems [5], but the first major European heat pump was installed in the Town Hall in Zurich in 1938 [6] (Figure 2.2).

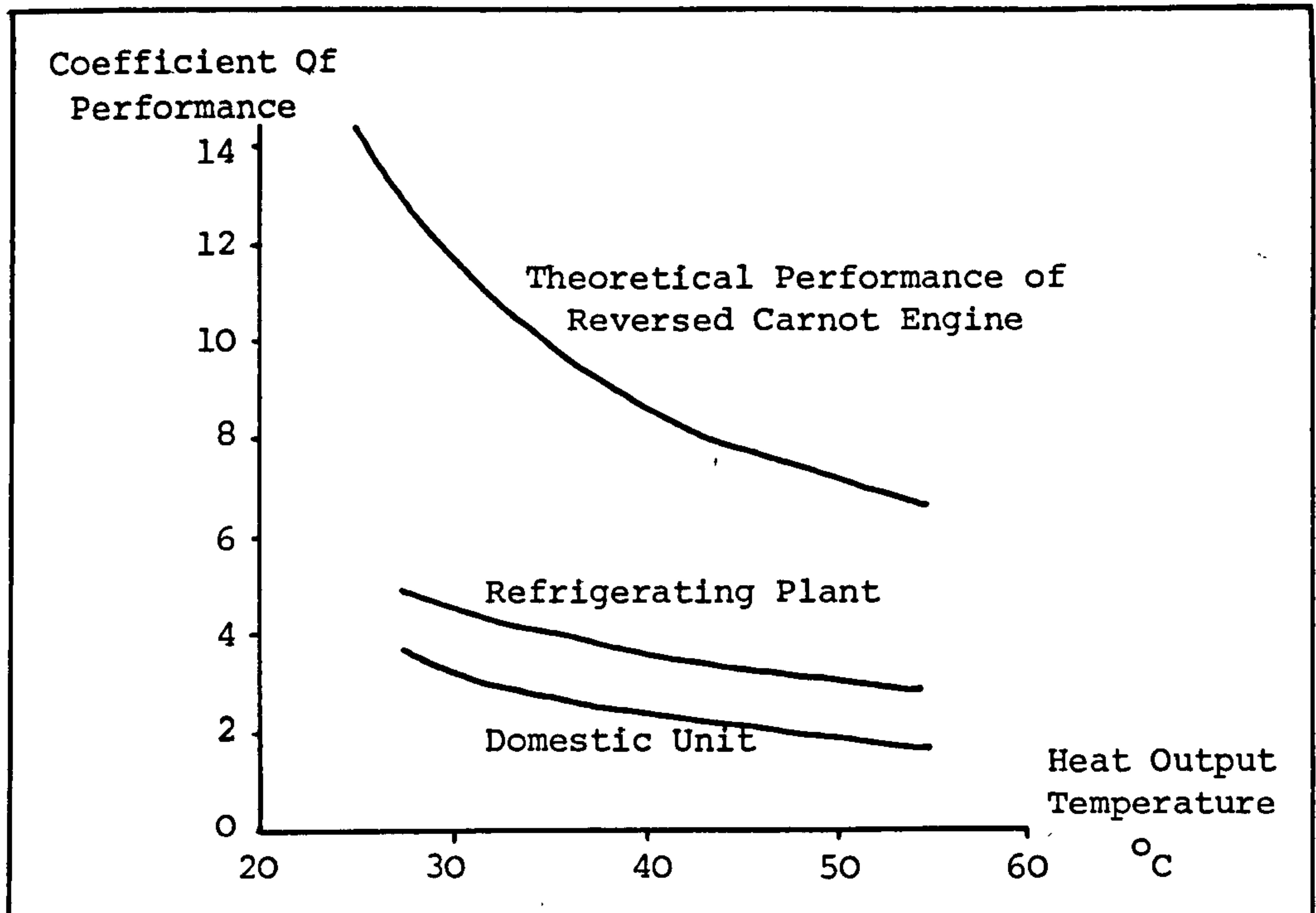


Figure 2.1 Heat Pump Performance for a Source Temperature of 4.4 °C. (Evaporating Temperature of -6.7 °C)

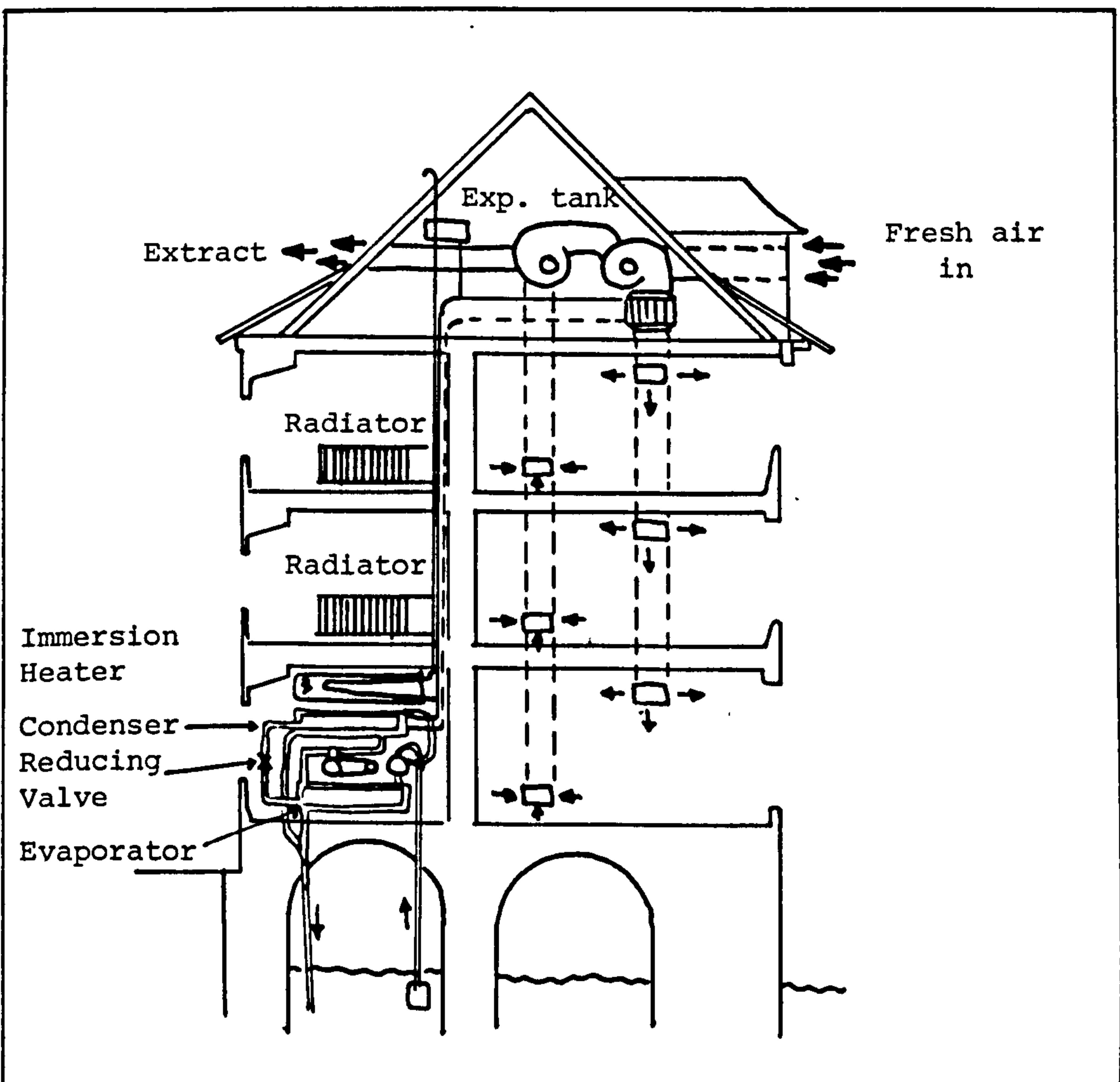


Figure 2.2 Schematic, showing layout of the Zurich Town Hall Heat Pump.

This used river water as its source of heat and incorporated a thermal storage vessel in the form of a calorifier which could also be heated electrically in times of peak demand. A rotary vane compressor was used, along with the refrigerant R12, and, from published results, the coefficient of performance was quoted to be between 2 and 3.5. Building on this success considerable interest grew up in Switzerland. A further scheme was undertaken for the Zurich public baths, the Lausanne artificial ice rink was connected to the town's water supply main, and another scheme for a silk manufacturer used water from Lake Constance as its heat source.

The first large scale heat pump to be built in Britain was built by Sumner [7] in Norwich in 1945 to heat the Norwich Corporation's electricity department (Figure 2.3).

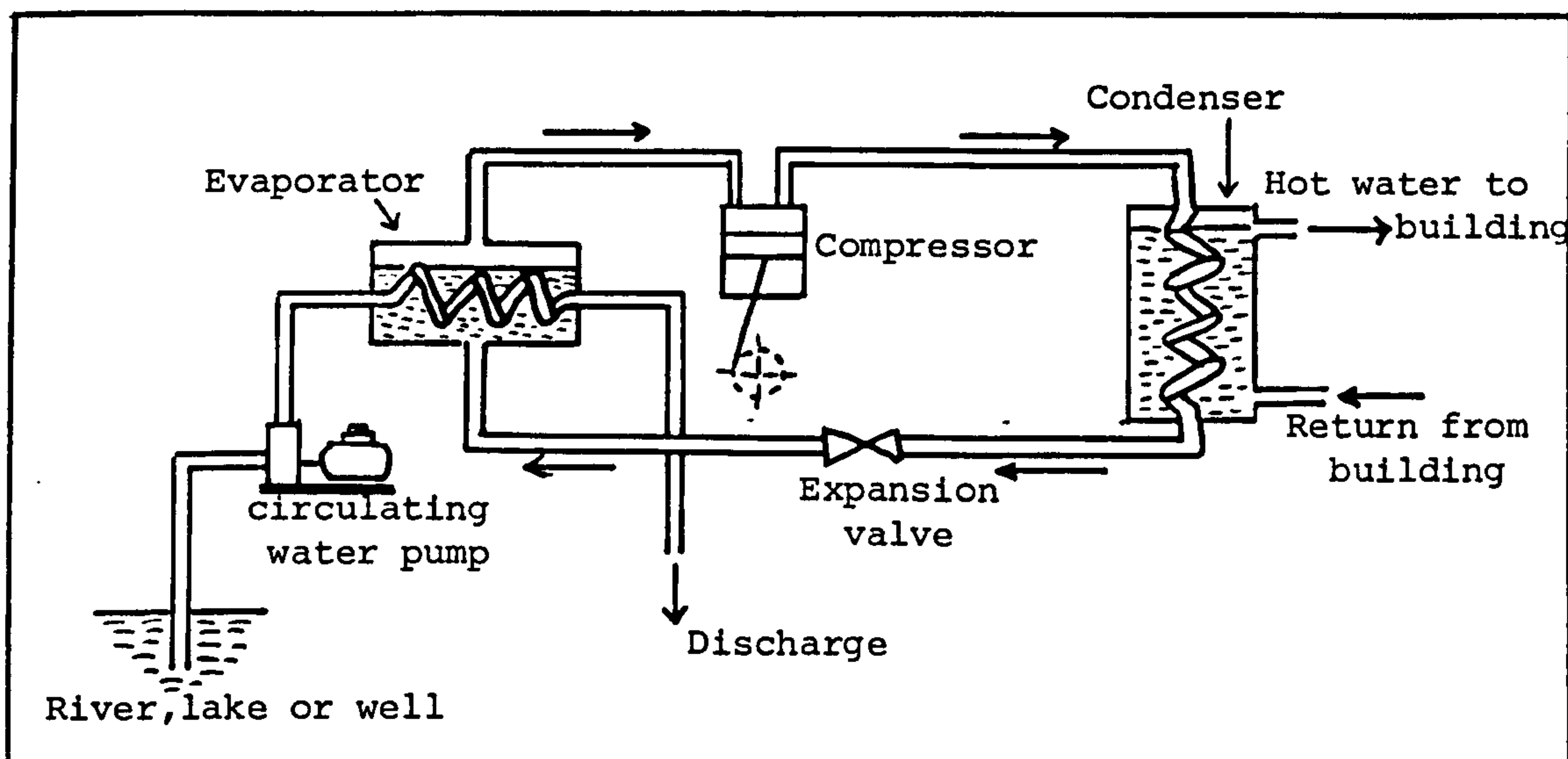
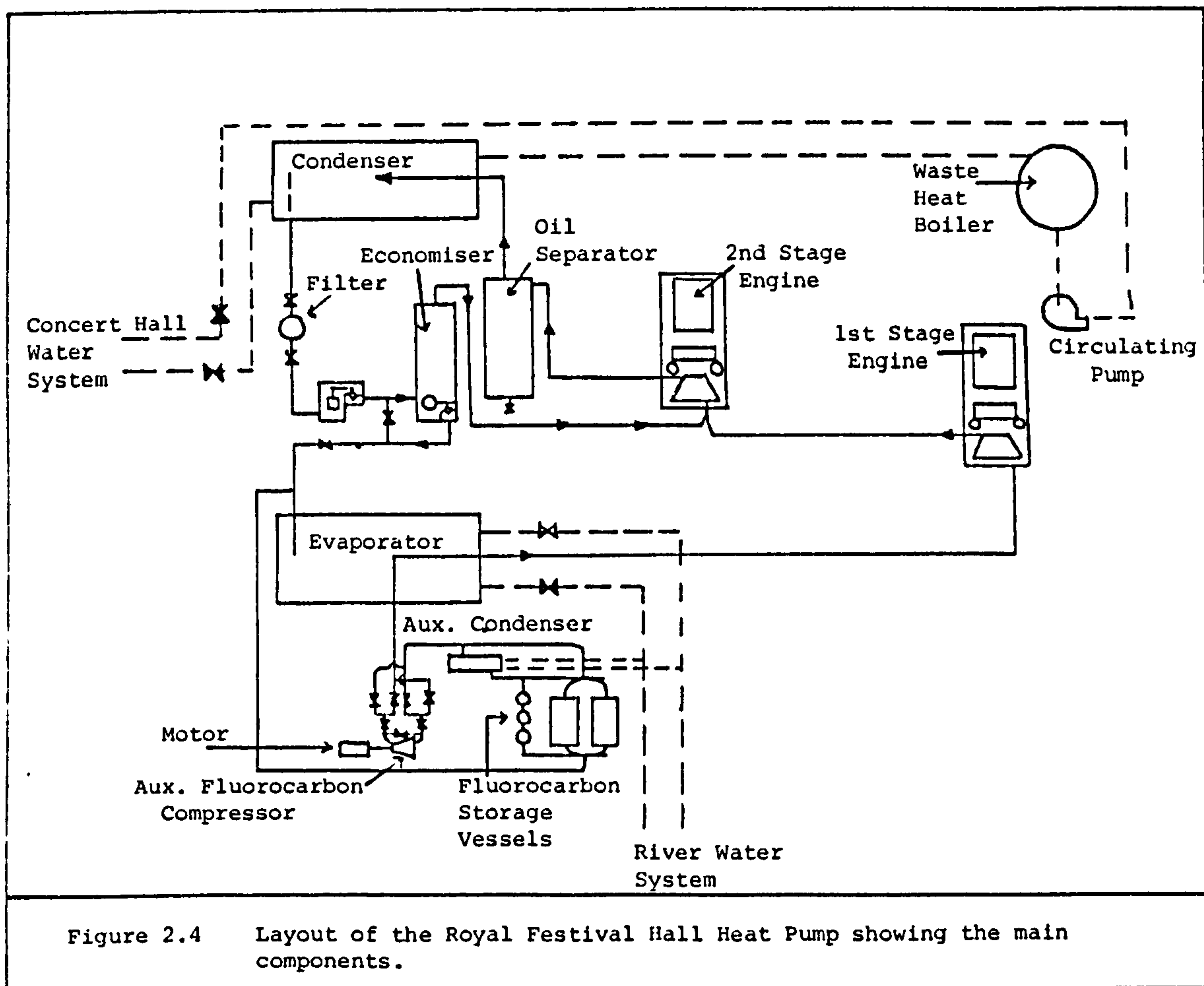


Figure 2.3 Schematic, showing elements of the Norwich heat pump

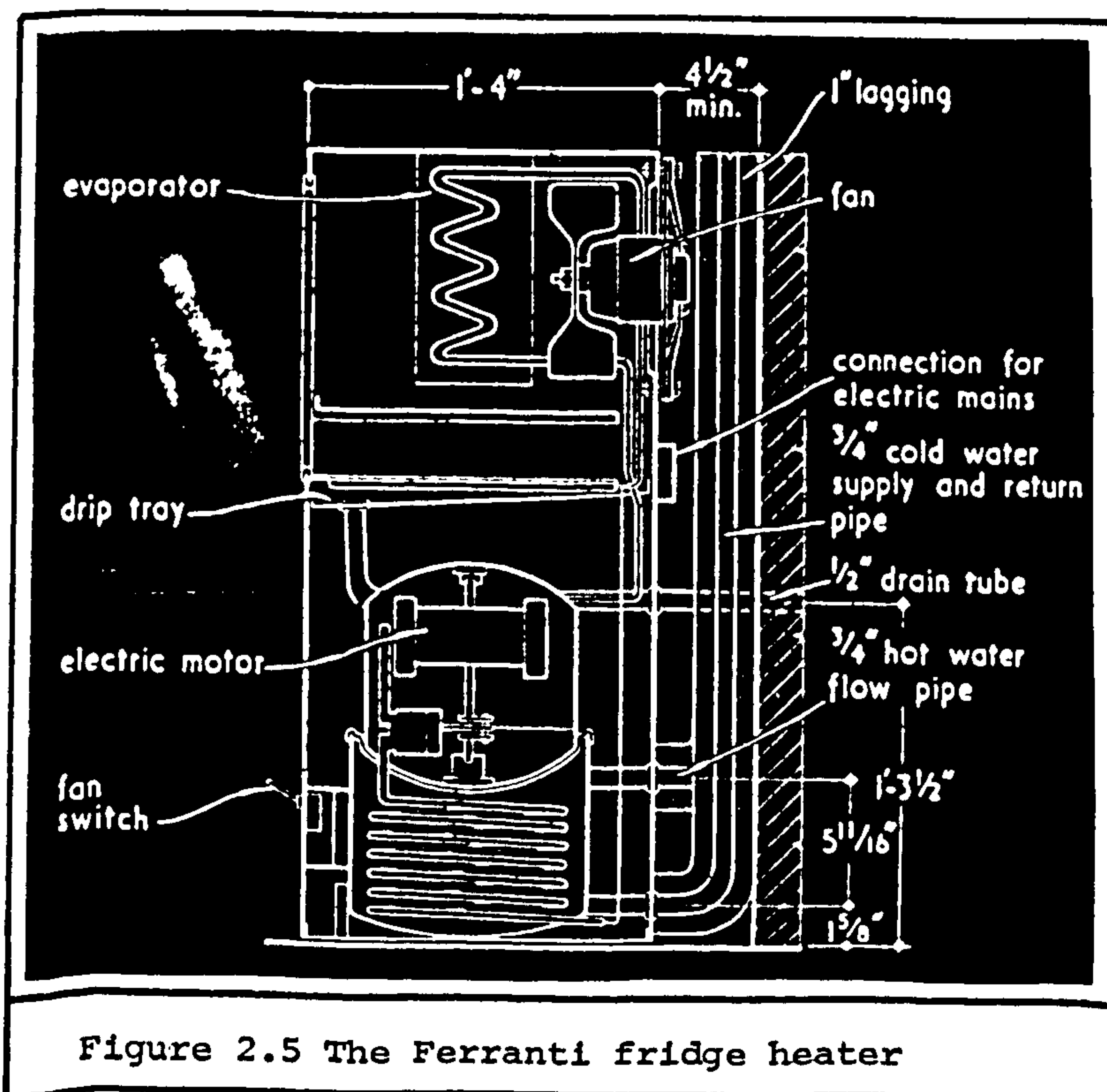
Here the heat was achieved using a sulphur dioxide refrigerant and a two-stage compressor, belt driven from a DC electric motor.

Of greater fame was the second installation - a heat pump at the Festival Hall in London in 1951 [8,9] (Figure 2.4). This had the dual role of providing heat in winter and cooling in summer, designed to give a maximum heat output of 2.6 MW. The river Thames provided the source of heat, and the high



speed centrifugal compressors were the converted superchargers of the Merlin aircraft engines which drove them. These caused a serious high frequency noise problem, but by far the major design fault was in fact, over design. The high level of sound insulation, and the heat given off by occupants meant that the heat pump unit was massively oversized and could therefore only be run for short periods at a time. This so called "failure" of heat pumps was the cause of much detrimental publicity for a number of years to come.

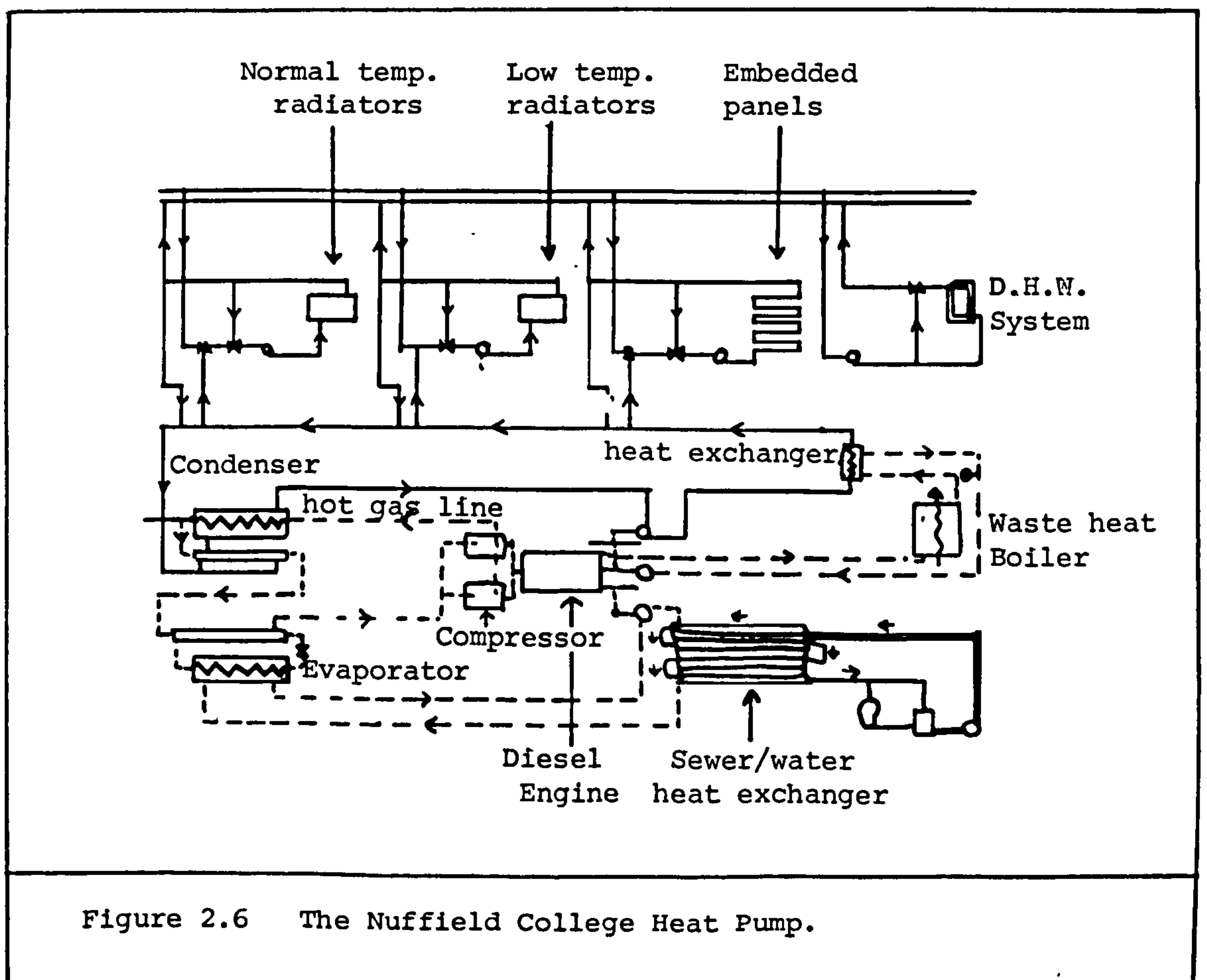
On the domestic scale, one of the first heat pump units was one designed and built by Sumner [10,11] in his own bungalow in Norwich, where it operated from 1952 to 1961. Originally the heat source employed was the air, but this was replaced later by ground coils, embedded approximately 1 metre below the surface of the ground. There were also several mass-produced heat pumps produced in the 1950's including an



+ The system is currently (1983) being rehabilitated, replacing the single diesel engine by two small spark ignition engines, running on natural gas.

air-to-air machine made by Lucas [12], and the Ferranti Fridge-heater [13] shown in Figure 2.5. on the previous page.

This was used to extract the heat from a larder or cold storage room, and this heat was then used for water heating. The only other heat pump of historical importance which must be mentioned is that installed at Nuffield College, Oxford in 1954 [14] (Figure 2.6). This system is still in existence today, although not still in operation⁺, and used sewage as its source of heat. A 31 kW diesel engine was used to drive the compressor, and an overall COP of around 4 was achieved. This system had the added advantage of being able to utilize the exhaust heat from the engine to boost the temperature of the water used to recover heat from the engine cooling jacket.



The market for heat pumps grew much faster in America during this period than in Britain, since the need there was for a dual purpose machine which could supply heating in winter and cooling in the summer. By 1963 76,000 units a year were being produced there and a healthy industry was beginning to emerge [15]. However, like all new products, there were some teething problems which resulted in heat pumps getting a reputation as being unreliable. This, together with an era of "cheap" and plentiful fuels, both in America and the UK, brought the heat pump industry in both countries to a virtual standstill.

2.2 More recent history - post oil crisis

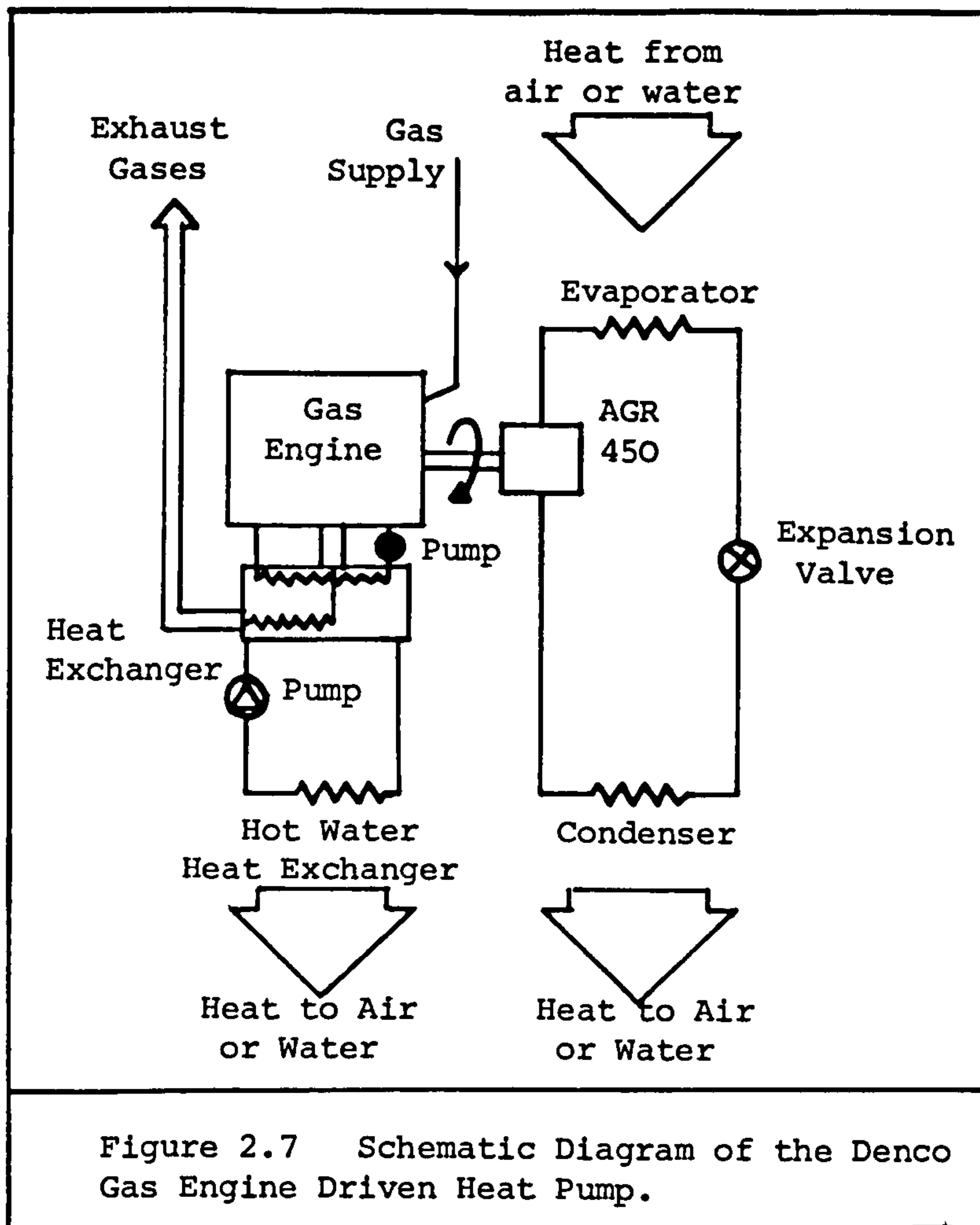
Fortunately for heat pumps, the 1973 "fuel crisis" came just in time and caused a revival in heat pump research and development to its present healthy state. However, most of the research to date has been confined to electric heat pumps which are now commercially well established in British and overseas markets. They are quiet, clean machines, easy to operate and relatively simple. However, in energy terms, electricity generation is still a wasteful process (typical efficiency of around 33%). This means that even with COP's of 3 or more, electric heat pumps can, at best, only supply the same amount of energy as was present in the original fuel.

With engine driven heat pumps however, this figure increases

dramatically, especially if use is made of the exhaust gases and the heat recovered from the engine cooling jacket. Against these advantages, problems of complexity, noise and unreliability must be taken into consideration.

Apart from the work described in Chapters 5 and 6 of this document, there are now several organisations and companies in Britain working on gas engine driven heat pumps [16], notably British Gas, the Department of Energy and three large manufacturers. By far the largest impetus has come from the car firm Ford, via their industrial engine division. They have converted their standard 2274E 1.6 litre engine for heat pump applications. The engine is modified to run on LPG or natural gas and 80-90% of the waste heat is recovered from the cooling water and exhaust. The oil capacity has been increased to lengthen the service intervals and potentially vulnerable areas such as bearings and valves have been improved. Ford have been carrying out extensive trials with a number of gas boards here and in the rest of Europe and have also been instrumental in promoting the use of gas engine driven heat pump systems through seminars, exhibitions, trade fairs etc. They have also actively encouraged other companies to enter the field, providing a network of service engineers and spares for back-up.

One of Ford's most successful link-ups has been with Denco Miller [17]. The Ford 2274E engine has been coupled to a Serck heat exchanger and a Denco AGR rotary compressor as shown in Figure 2.7 overleaf.



The prototype gave an average output of 120kW with a P.E.R of 1.2 for over 2000 hours of a test programme and the production model is now in full swing. Ruhrgas, have also conducted extensive reliability tests with the Ford Engine, including running one engine at full throttle for 1200 hours with only routine maintenance.

So far over 30 European manufacturers have used Ford engines to power their heat pumps, with four German manufacturers in series production. Now other European car firms are also showing an interest namely Renault, and Fiat, who started off in the market with their Totem - total energy module as shown in Figure 2.8, but now look extremely likely to enter the heat pump market in the near future [18].

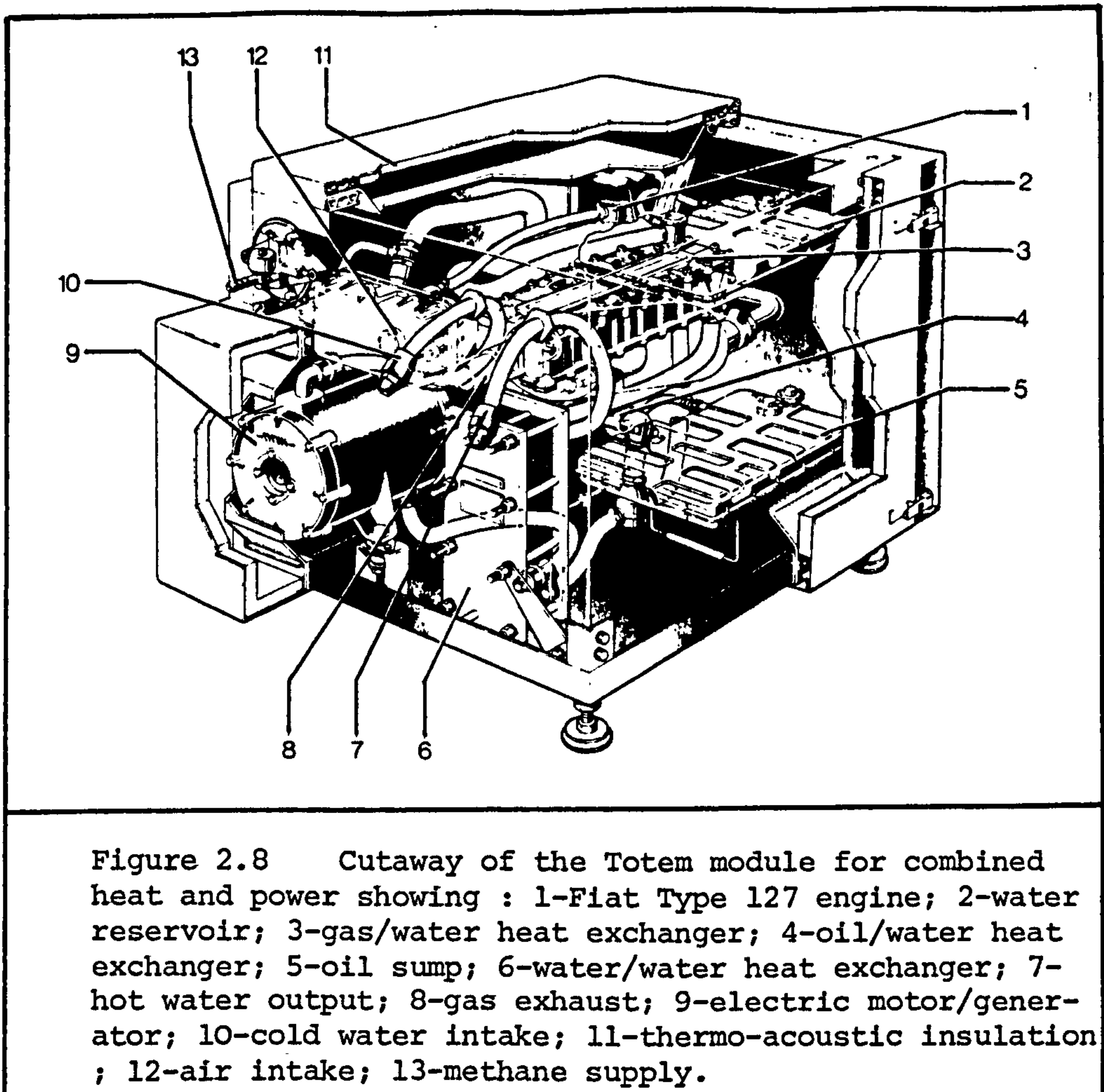


Figure 2.8 Cutaway of the Totem module for combined heat and power showing : 1-Fiat Type 127 engine; 2-water reservoir; 3-gas/water heat exchanger; 4-oil/water heat exchanger; 5-oil sump; 6-water/water heat exchanger; 7-hot water output; 8-gas exhaust; 9-electric motor/generator; 10-cold water intake; 11-thermo-acoustic insulation ; 12-air intake; 13-methane supply.

Volkswagen have a Research Development centre at Wolsburg in West Germany, and have been working on heat pumps for some time but only unveiled their package early this year, based on the Golf car engine. They carried out an extensive market survey which showed that over 6 million oil fired boilers would be coming up for renewal between 1983 and 1986. Aiming for a large replacement market they have designed their package so that all connections to the oil boiler can remain in the same place, and the same pipework can be utilised. Installation can then be carried out in under 12 hours.

West Germany suffers a severe winter climate with temperatures frequently below -15°C . Like Britain therefore, their need is for a heating only heat pump (as against the

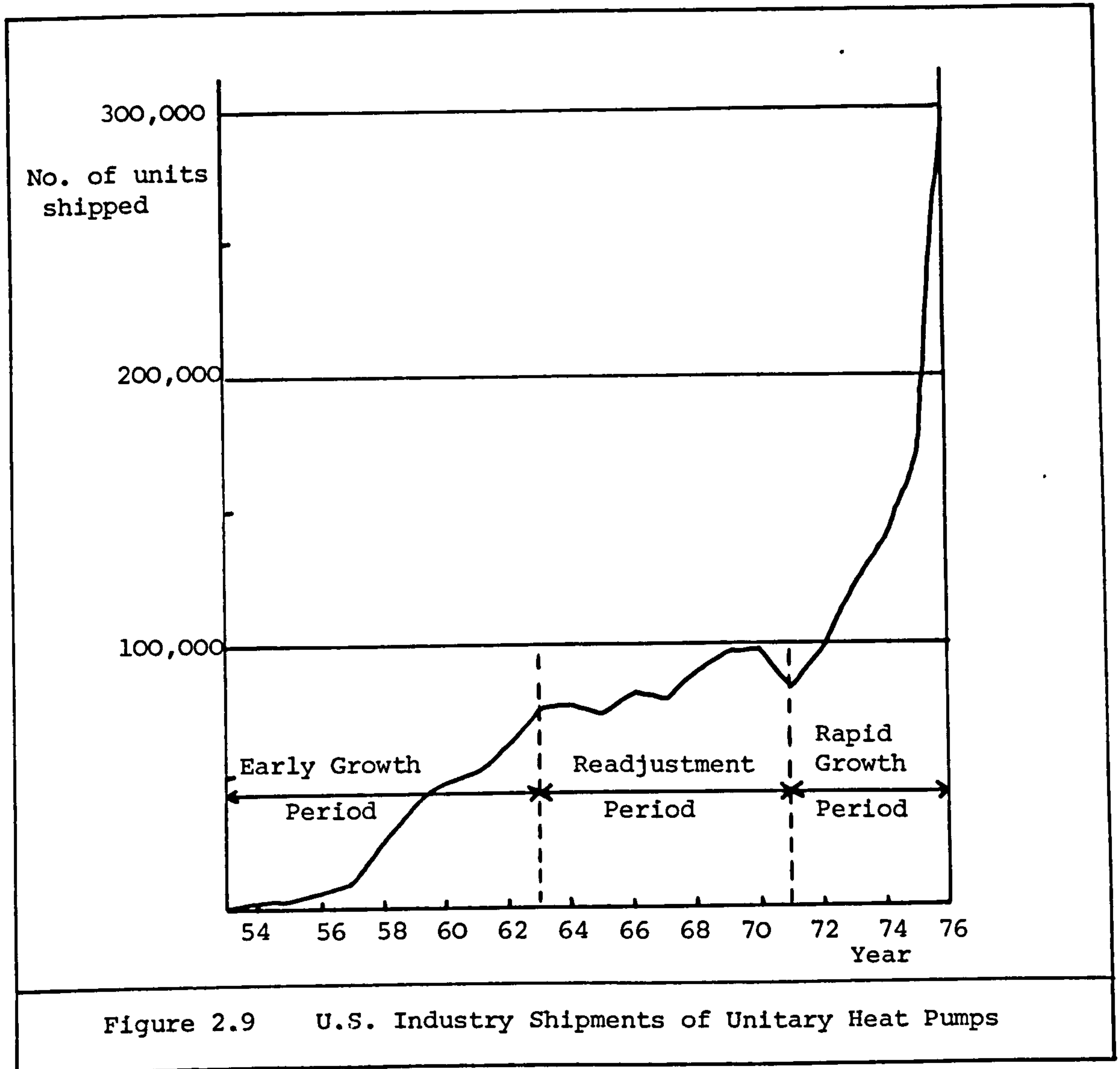
U.S.A). [19]. Because German weather can be very severe however, they mostly adopt a bivalent heating system. Since the efficiency of any heat pump drops as the ambient temperature drops, a point is reached where the heat pump cannot effectively, or efficiently, produce enough heat to meet the demand. At this point some sort of supplementary heating is used, normally electric. During prolonged periods of extreme cold weather such a system can be very expensive and it can cause problems for the electricity supply industry, adding to peak loading. An alternative method would be to use a separate fuel fired boiler to supply all the heating above the balance point. This has the effect that both systems are then running at maximum efficiency and also the peak time electricity demand is reduced. In Germany, heat pumps used in this manner are an acknowledged form of heating in residential, domestic and industrial premises and their number is likely to reach the one million mark by 1990. Generally air/water/soil to water machines are used in the domestic sector in Germany, and, since underfloor heating is popular, the output temperature of water required is not too high.

In Sweden [20], despite its bitterly cold climate, the heat pump market is for reversible machines, since due to extremely high levels of insulation, summer cooling is also required. Air to air machines are the most popular and most of these adopt the method of allowing the warm exhaust air from the building to be discharged over the outdoor evaporator coil of the heat pump, to improve the COP and

reduce the need for defrost. In Sweden and Germany there are grants available as incentives for heat pump manufacture and installation [21]. Sweden currently has some 100 heat pump projects underway and Germany 50 projects.

In the U.S.A, the market for heat pumps developed much faster than in Britain, and a mass market quickly built up in the 1950's, with the emphasis on summer cooling. As was explained in the previous section, due to the unreliability of these machines and an ensuing era of cheap energy, a decline followed. However, this decline in America was short lived and again due to the "energy crisis" of the 70's, heat pumps found themselves in the foreground once more. As can be seen in Figure 2.9 overleaf, sales of small packaged units had topped 300,000 by 1976 and today there are well over 2 million, although here the market is almost exclusively for electric systems.

Japan and Russia [22] are also showing renewed interest in heat pump research and a number of experimental installations have been built in these two countries. Indeed progress in heat pump research is now being monitored worldwide and takes place in many other countries e.g. Australia, Austria, Belgium, Denmark, Finland, Italy, New Zealand, Norway and Switzerland [23]. The monitoring organisations include the World Energy Conference (WEC) [24], International Energy Agency (IEA) [25], and the Scientific Technical Research Committee of the European Community (REC-CREST) [26].



CHAPTER 3 WHAT IS A HEAT PUMP?

The aim of this chapter is to look at heat pumps and their components in more detail.

There are two basic cycles which go together to make up any heat pump: the power or engine cycle, and the refrigeration cycle. There are many variations and combinations of both and this chapter examines the theoretical principles behind them, before concentrating on the vapour compression heat pump, chosen as the basis for the experimental unit described in more detail in Chapters 5, 6 and 7.

There are also numerous other choices for the designer of any heat pump e.g. a choice of working fluid, of heat source and of compressor. This Chapter discusses the relative advantages or disadvantages of each.

3.1 Heat Pump Cycles

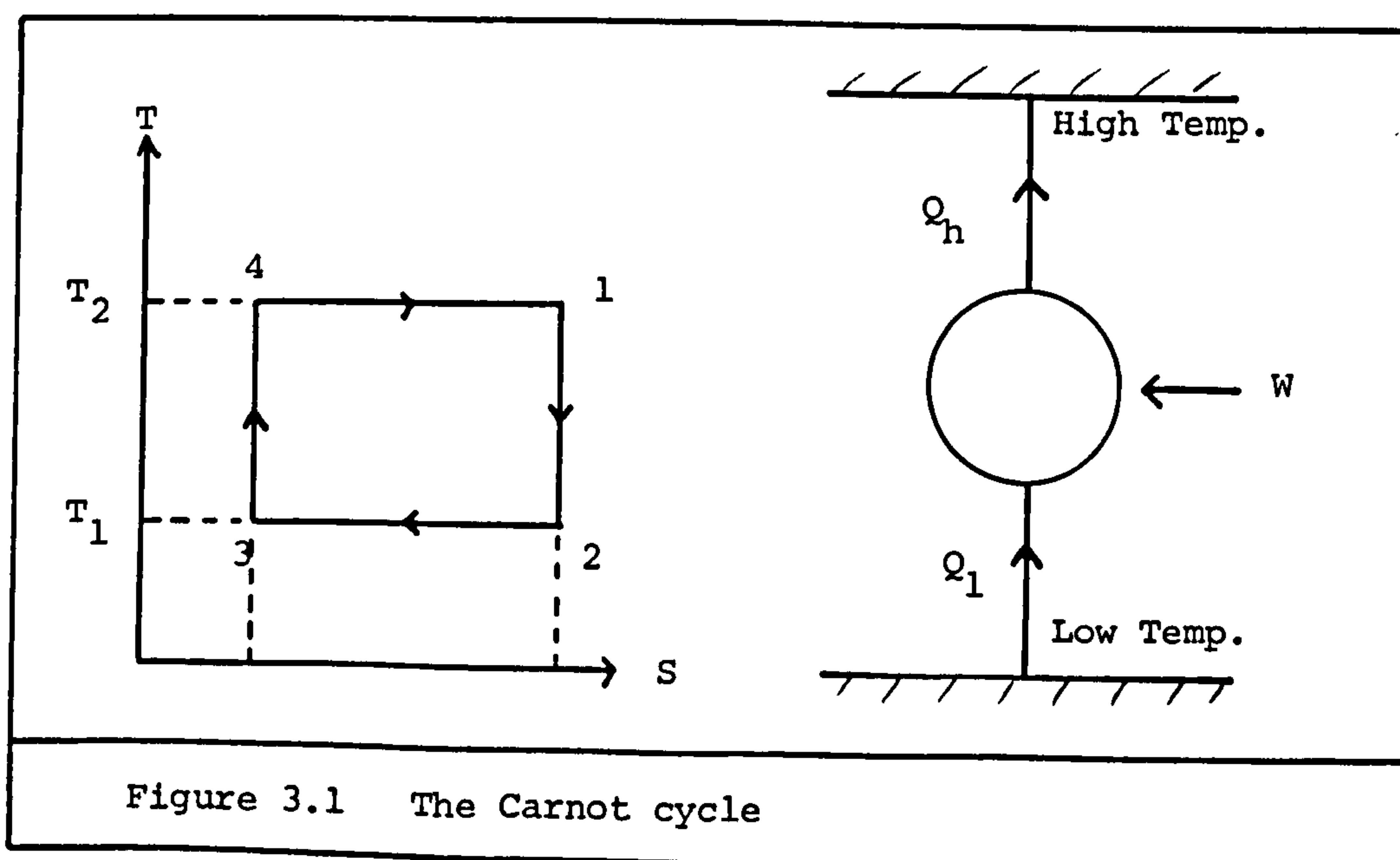


Figure 3.1 The Carnot cycle

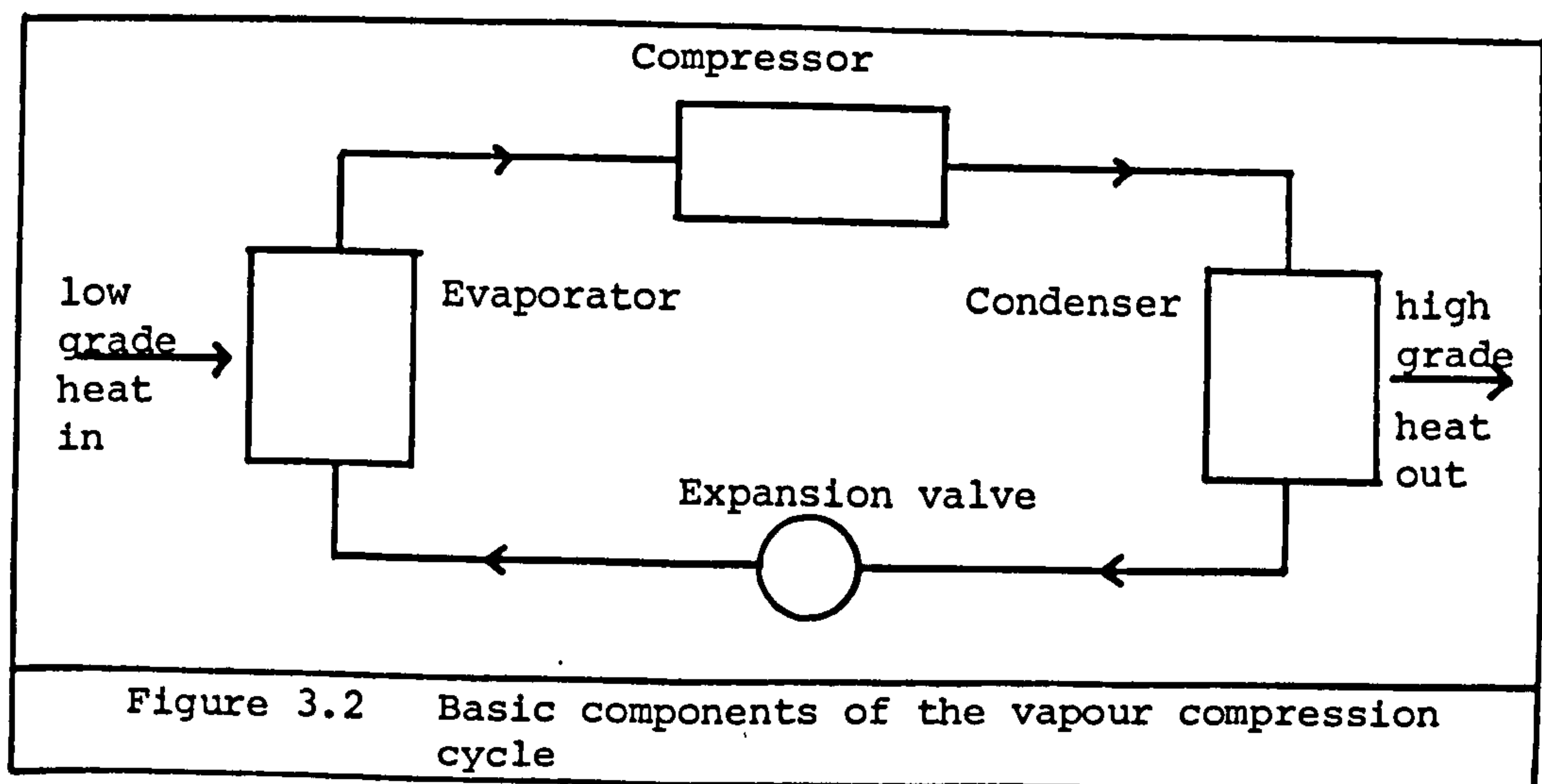
Before considering the various heat pump and power cycles, we

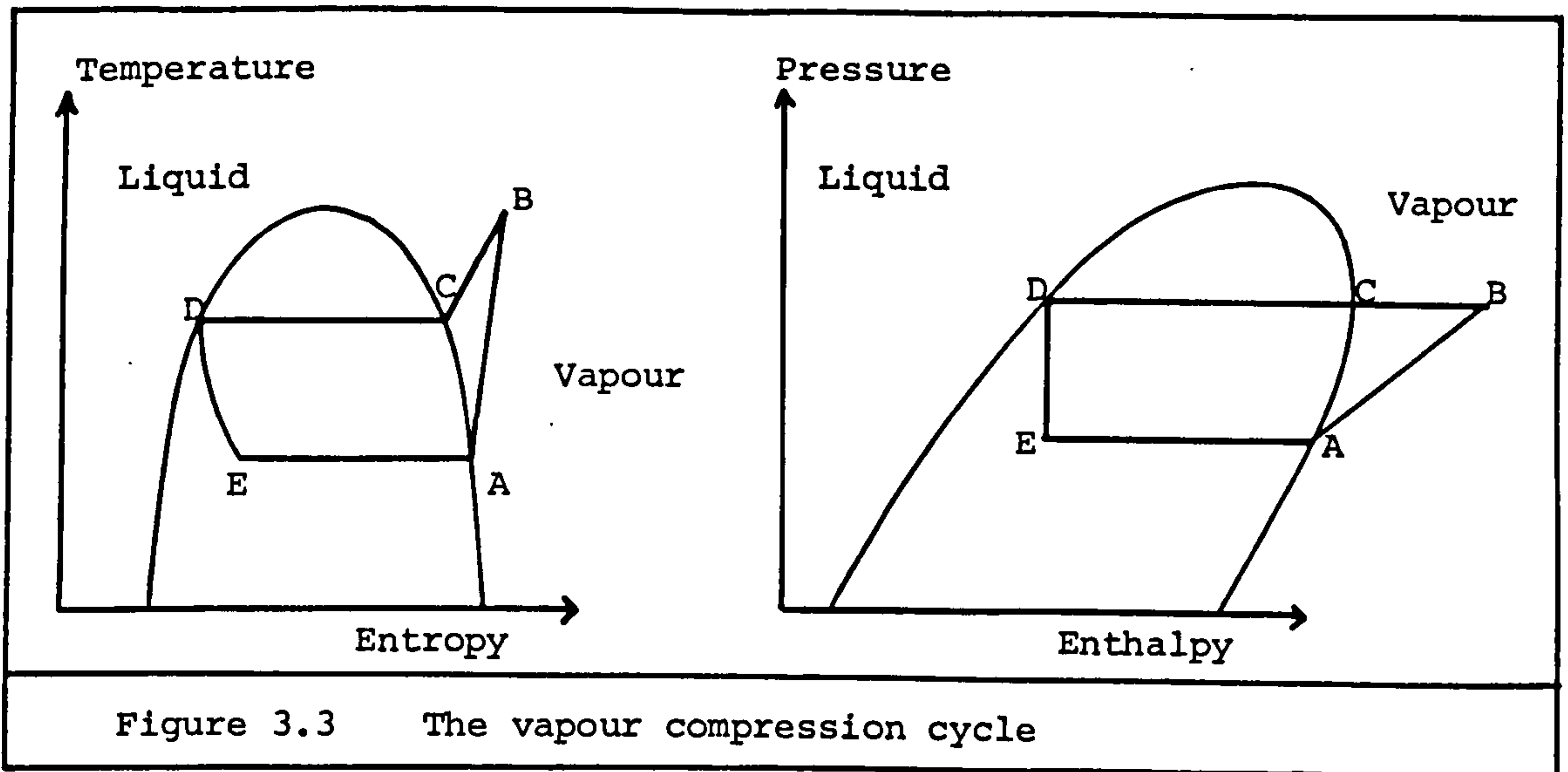
must first be familiar with the Carnot cycle, an ideal theoretical cycle, which describes the thermodynamic limit for heat pump efficiency.(Figure 3.1)

In 1824 Carnot showed that the most efficient possible cycle is one in which all the heat supplied is at one fixed temperature and all the rejected heat is rejected at a lower fixed temperature. Thus a heat pump is a heat engine in reverse and the cycle consists of 2 isothermal processes, connected by 2 adiabatic processes. The adiabatic process is also isentropic, and the cycle is independant of the working substance used. [1]

3.1.1 Vapour Compression Cycle

The vapour compression, or reversed Rankine cycle is the most widely used cycle for the majority of domestic and industrial refrigeration plant. The basic components of a vapour compression heat pump are shown in Figure 3.2 below and the cycle itself is illustrated on both temperature/entropy and pressure/enthalpy diagrams in Figure 3.3 overleaf.

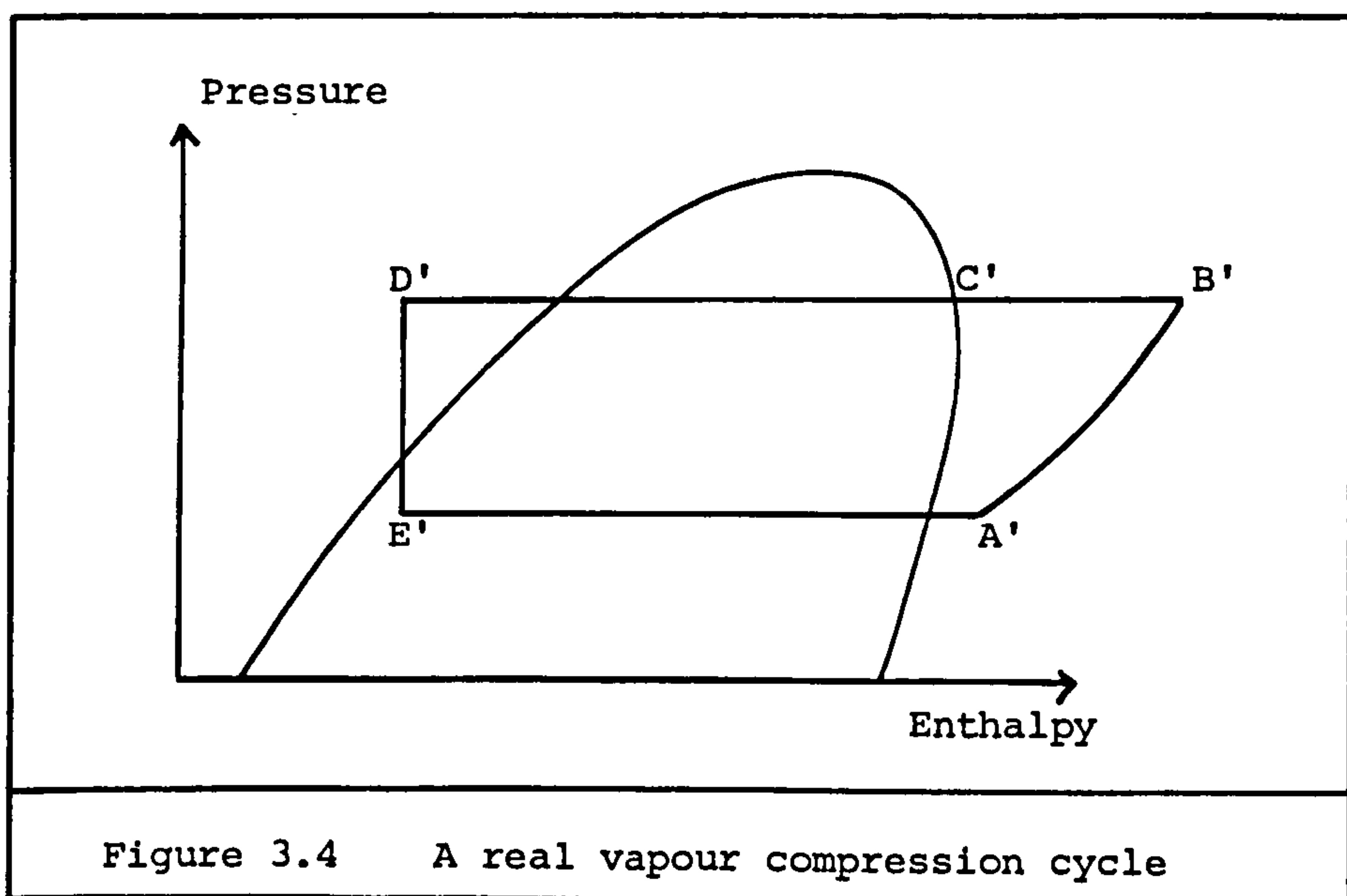




The working fluid or refrigerant, operates on a cycle approximating to the theoretical reversed Carnot cycle. Refrigerant vapour is compressed from point A to point B. Since liquid refrigerant could damage the valves of a compressor all compression must be carried out using dry vapour. The vapour at point B is now said to be superheated and must be cooled at constant pressure until point C is reached and it begins to condense. Condensation continues, at constant temperature from C to D until no vapour is left, and heat is released. The liquid refrigerant then passes through an expansion valve where it undergoes adiabatic expansion and its pressure is reduced (D-E). The liquid then enters the second heat exchanger, the evaporator. From point E to A, heat is absorbed at low temperature, and evaporation takes place at constant pressure and temperature resulting in the dry vapour ready to enter the compressor and repeat the cycle.

Although the cycle described overleaf takes account of the need for dry compression it is still an ideal cycle and assumes that the various components are all 100% efficient. Ambrose (2) has made a comparison of theoretical and actual cycles for various small electric heat pumps showing an overall C.O.P. of about 13% of the ideal Carnot values. As an example let us consider some of the problems of a realistic working cycle.

Because the compressor can only operate with dry vapour and to prevent damage to the valves, a certain amount of superheat takes place which can be seen in Figure 3.4.



Thus the refrigerant now enters the compressor at state A' rather than A. To accommodate the less dense vapour at the same mass flow rate, the size of the compressor, and the discharge temperature, must be increased. There are three measures of compressor efficiency: Isentropic efficiency (70-80% for a practical reciprocating compressor) occurs because of the heat transfer between the working fluid and the

compressor, and, because of the irreversibility of the flow through the compressor, the enthalpy is increased by a greater amount than is necessary. The mechanical efficiency is a measure of how much of the work applied to the shaft is delivered to the working fluid, and finally the volumetric efficiency is a measure of how efficiently the refrigerant passes through it. Typical values for both of these efficiencies are 95%.

The other areas of loss of efficiency in the system are pressure losses in the pipes, finite temperature differences in the heat exchangers, and any fan power necessary. Yet another change to the ideal cycle, although this time not affecting the C.O.P. is the desirability to subcool to point 0^1 in Figure 3.4. The most elegant solution to this is to use a super heater with a heat exchanger or 'intercooler'.

In larger systems the performance may be improved by the use of two or more compressors in series or in parallel. A two stage system is illustrated in Figure 3.5 overleaf. The dotted lines represent a single stage cycle between the same pressure and on comparing the two it can be seen that there are reductions in both subcooling and superheating, leading to an increased efficiency. This can be seen in Figure 3.5(b) based on calculations by Durnimal [4].

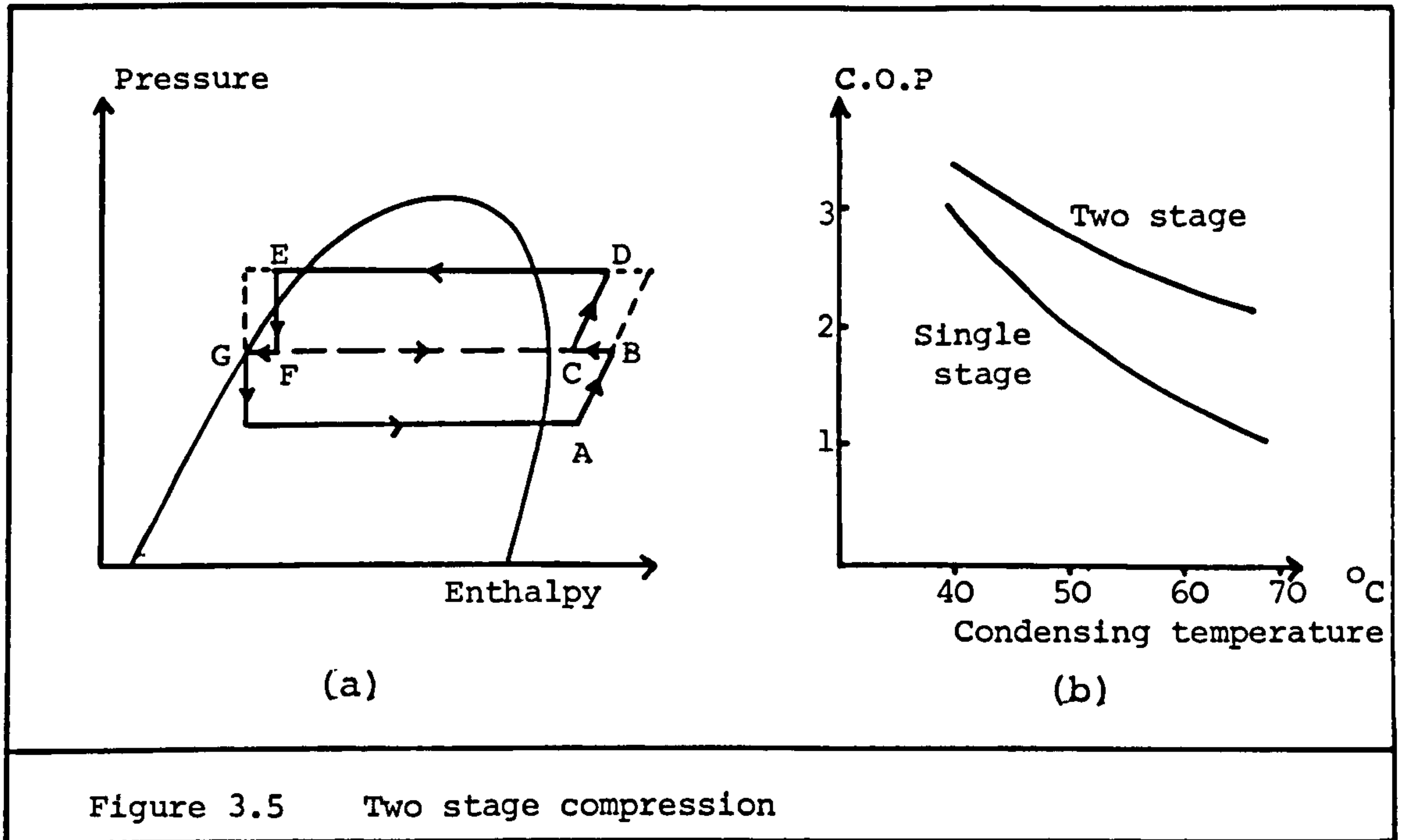


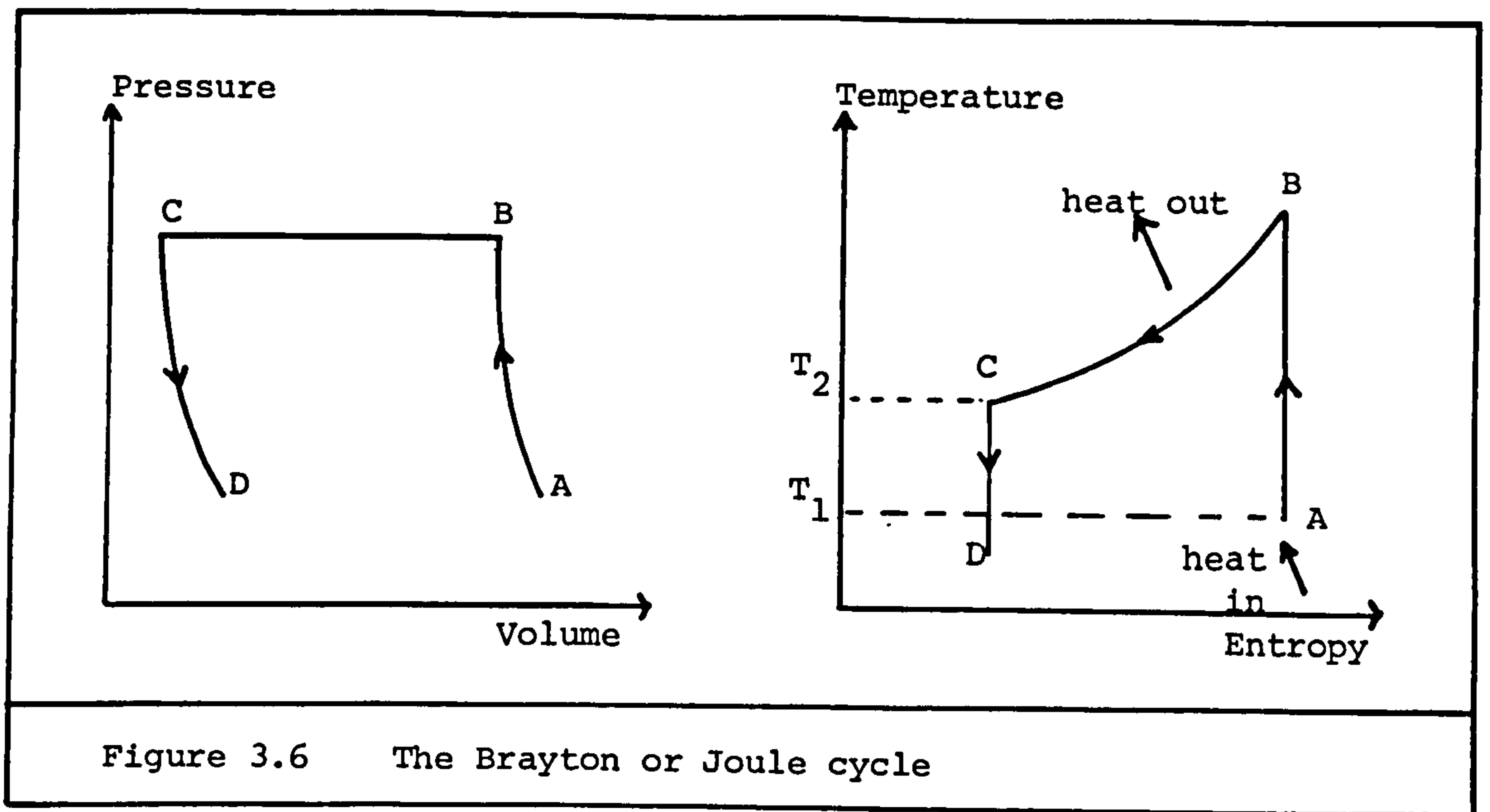
Figure 3.5 Two stage compression

3.1.2 Other Cycles

There are a number of other interesting heat pump cycles worthy of mentioning here, although not as widely used as the vapour compression cycle.

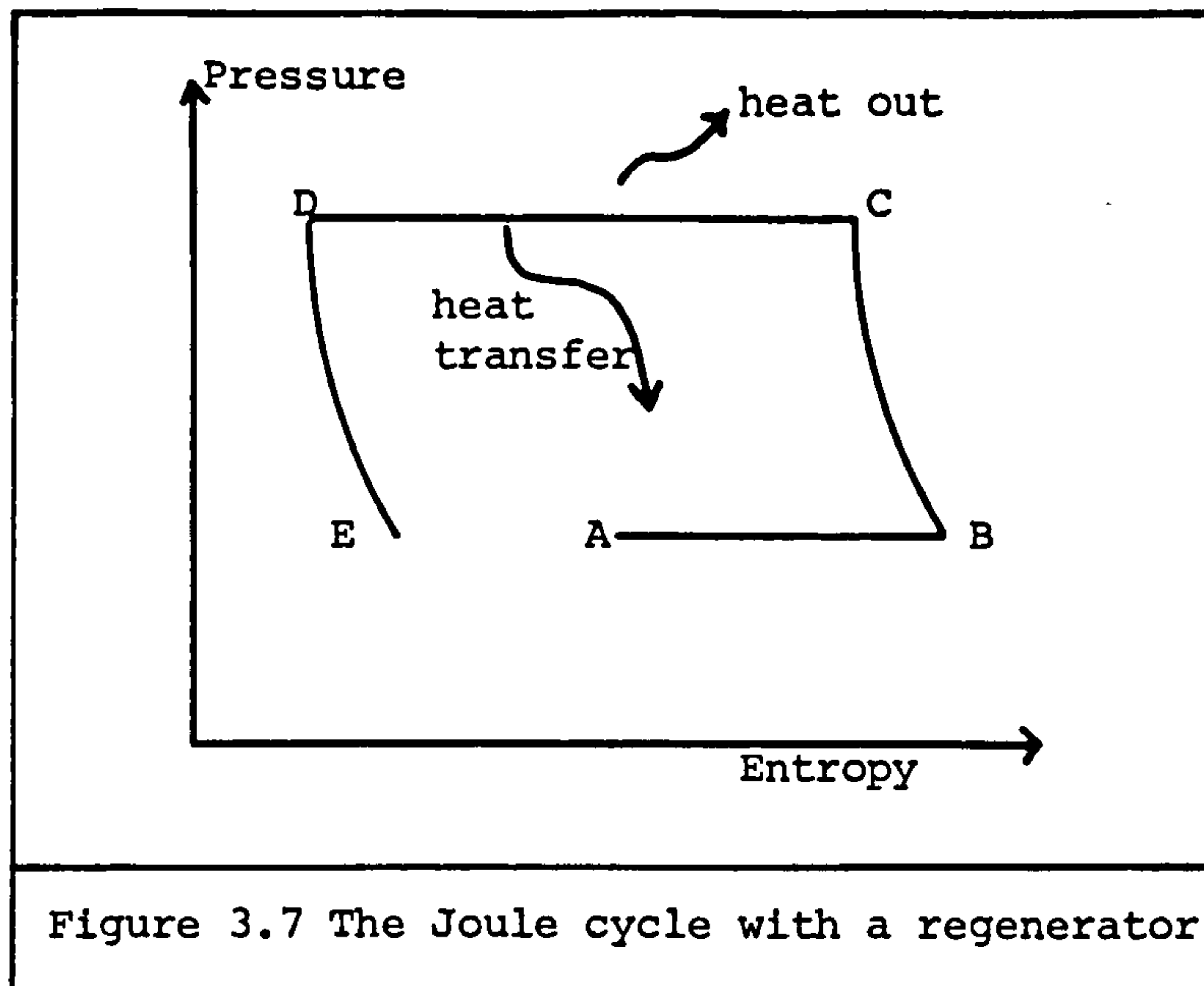
The Brayton or Joule cycle [5] is commonly used for aircraft airconditioning and is simply the reverse of the power cycle used by the gas turbine engine. The advantage of this is that it is an open cycle, using air as its working fluid, and only needs one heat exchanger - all factors which lead to low capital cost. The C.O.P. achieved, however, is low.

As can be seen in Figure 3.6 ambient air is drawn in at point A, and compressed isentropically, increasing its temperature to point B. From B to C it is cooled by delivering heat (via



a heat exchanger). From C to D the air is expanded back to atmospheric pressure, and finally exhausted at point D. Work resulting from the expansion process provides some of the work needed for compression. During the expansion process the air in the expander gets so cold (-20° on a 0°C day) that there is a considerable icing up problem and to overcome this in practice a closed cycle would have to be used. In such a system air from point D passes through a heat exchanger, receives heat from the outside air and is re-used at point A. Although the icing problem would then be easier to deal with, being on the outside surface of the expander, the C.O.P. would be greatly reduced.

To obtain a worthwhile C.O.P. the system must use a regenerator as is shown in Figure 3.7[6,7]. Heat is then transferred via a heat exchanger from the previously cooled high pressure air to the fresh air intake. This increased complexity however, does not appear to be cost-effective,

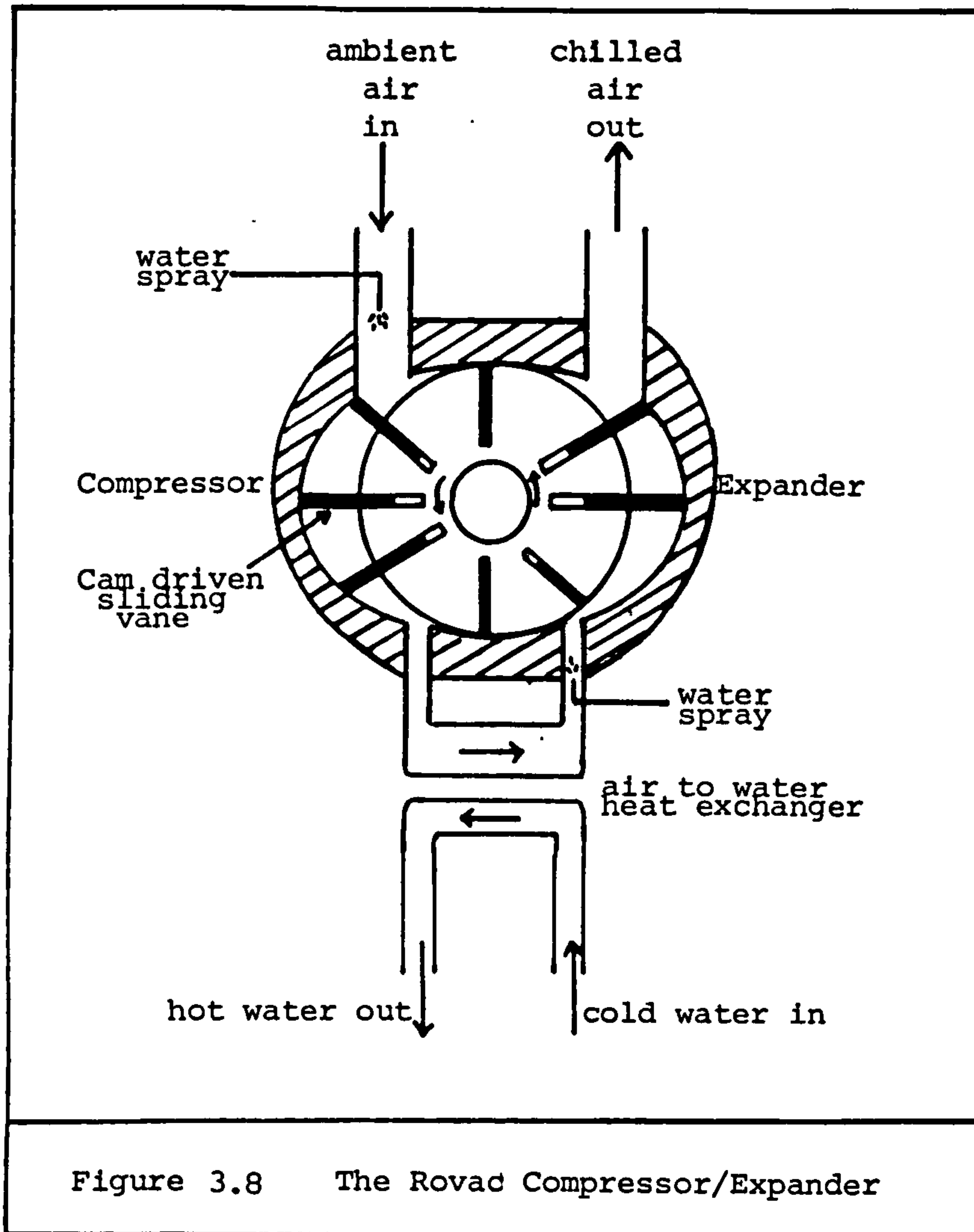


especially when the Edwards cycle [8] is considered, which is a slight modification to the Brayton cycle and seems much more promising, with a C.O.P. competitive with the Rankine cycle.

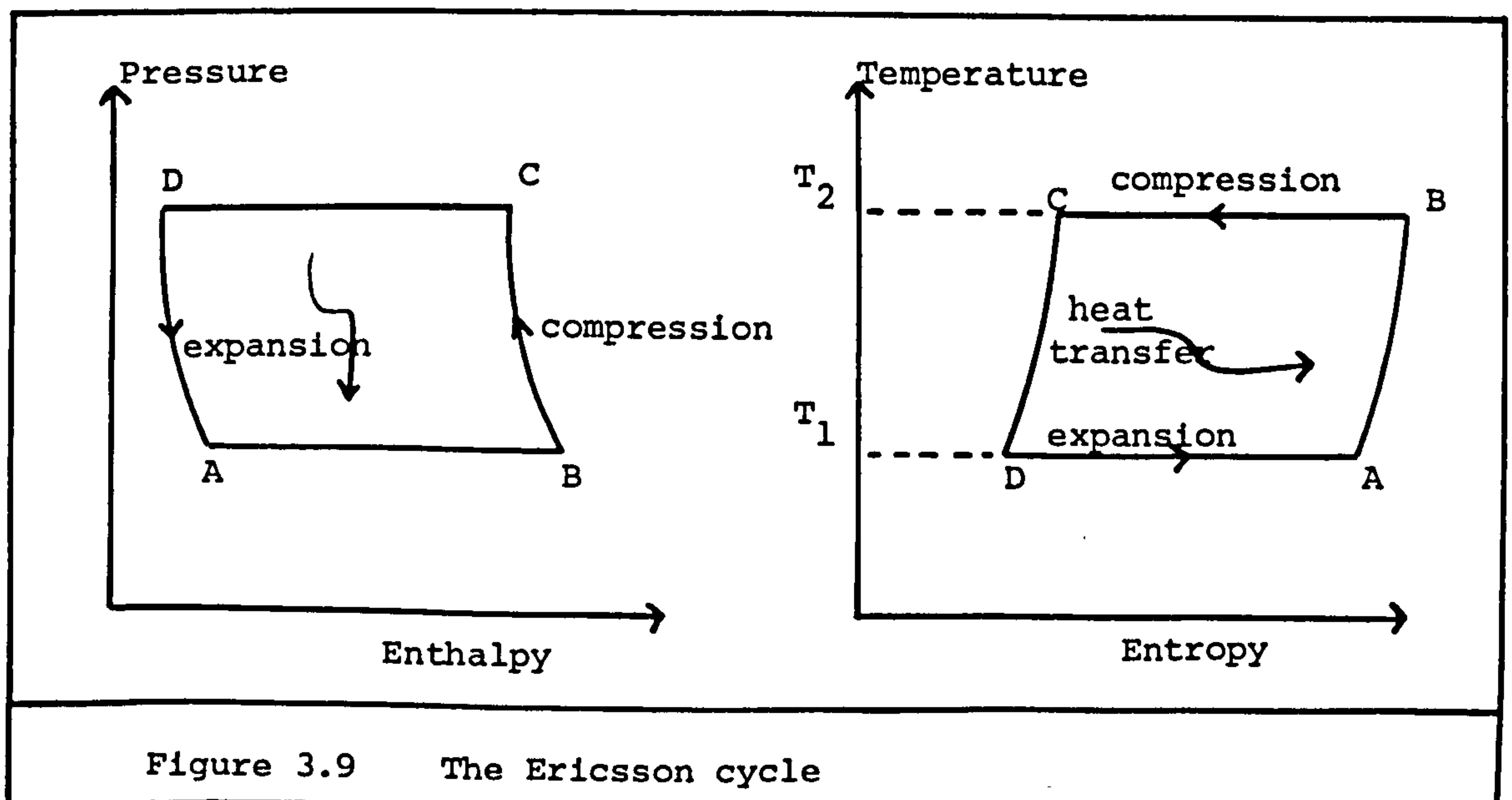
The difference between the two lies in the choice of working fluid. Instead of using air, the Edwards cycle uses a mixture of air and water. As Figure 3.8 shows the water is added as a fine spray prior to compression and reduces the work of the compressor, for a given pressure rise, by cooling the air. Similarly, another water spray is used before the expansion process, and, as the water freezes, the work of expansion increases. To prevent the blades from icing up a combined rotary vane expander and compressor is used and any ice crystals are expelled with the exhausted air.

Both the cycle and the compressor/expander were invented by T.C. Edwards [9,10] of the Rovac Corporation and envisaged by him for use in car air-conditioning and solar assisted heat

pumps.



Both Ericsson and Stirling cycle heat pumps use a gas (usually air) as the working fluid and ideally they have Carnot C.O.P.'s. The Ericsson cycle is shown in Figure 3.9.



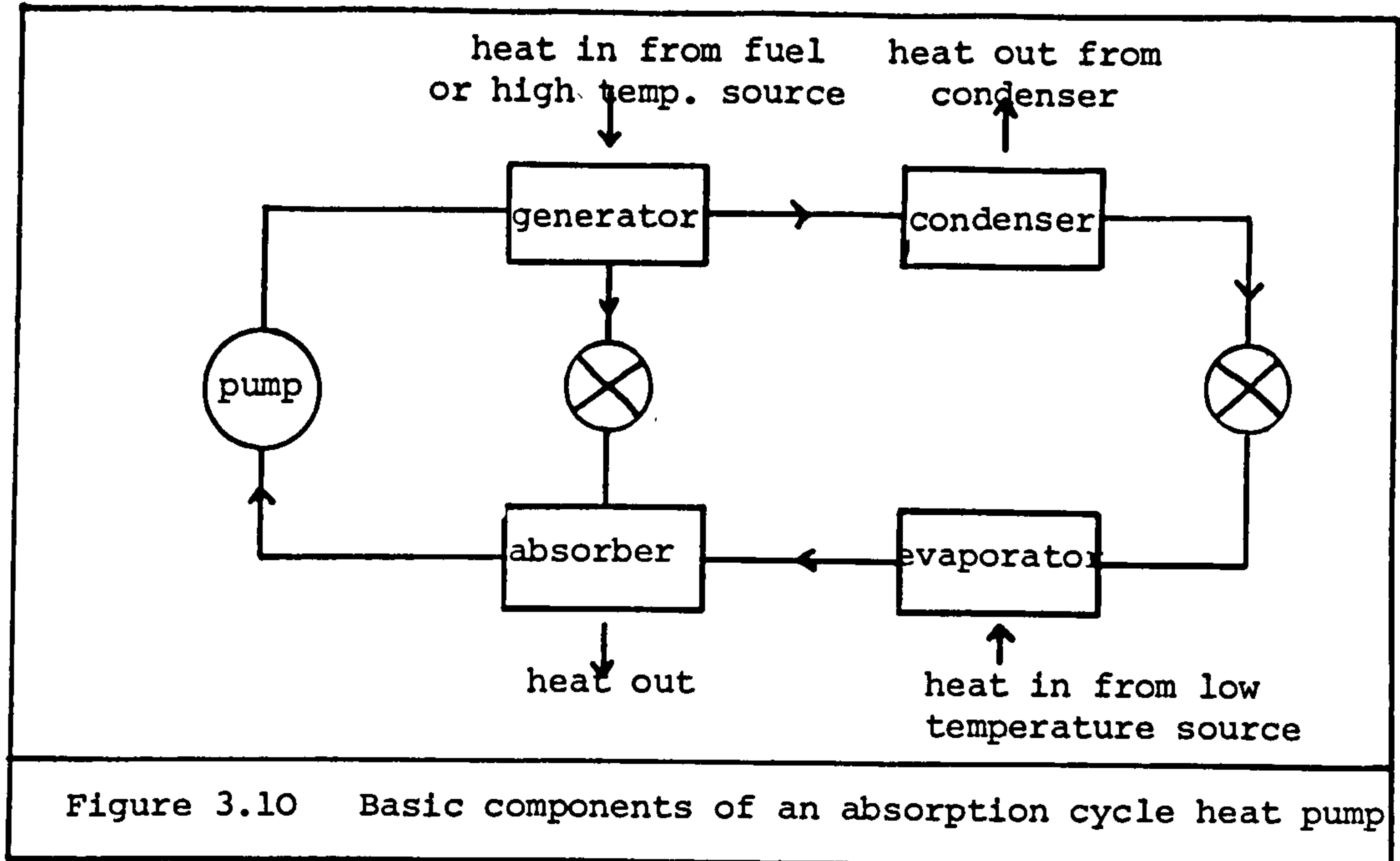
From B to C the working gas is compressed isothermally and reversibly. From C to D the gas is cooled in a regenerator and then expanded isothermally and reversibly from D to A. The regenerator supplies heat to raise the temperature of the gas from A to B. Thus since all the heat is supplied and rejected isothermally, the ideal efficiency between the same temperature limits is Carnot. Results by Benson [11] in the United States suggest a slightly better performance than Rankine.

The Stirling cycle is similar to that of the Ericsson cycle except that the process of regeneration takes place at constant volume rather than constant pressure.

3.1.3 Absorption Cycle

Before concluding this section on heat pump cycles some mention must be made of the absorption cycle which is a well established method of refrigeration and air conditioning. The cycle was first invented by Sir John Leslie in 1810 and since that time has been put to many uses ranging from small scale domestic units to the utilisation of waste heat in larger industrial units.

The basic components of an absorption cycle heat pump can be seen in Fig. 3.10. The evaporator and condenser on the right of the figure operate in exactly the same way as they do in a vapour compression system, but here the compressor is replaced by a secondary circuit consisting of an absorber, a generator and a pump. Low pressure refrigerant vapour from



the evaporator is dissolved and condensed in a secondary fluid called the absorber and the heat evolved is ejected. The resulting highly concentrated solution is then passed to a generator where it is heated by, for example, combustion of a fuel. This stage then separates off the, by now high pressure, refrigerant vapour from the absorbent, which returns to the absorber. The refrigerant vapour then passes to the condenser and the remainder of the cycle is as in the vapour compression cycle.

If we consider the absorption heat pump as a Carnot cycle heat engine driving a reverse Carnot cycle heat pump then the efficiency of the single stage absorption heat pump can be calculated as

$$\text{COP}_H = \frac{T_2}{T_2 - T_1} \frac{(1 - T_1 T_a)}{T_2 T_g}$$

where T_a is the absorber temperature, and

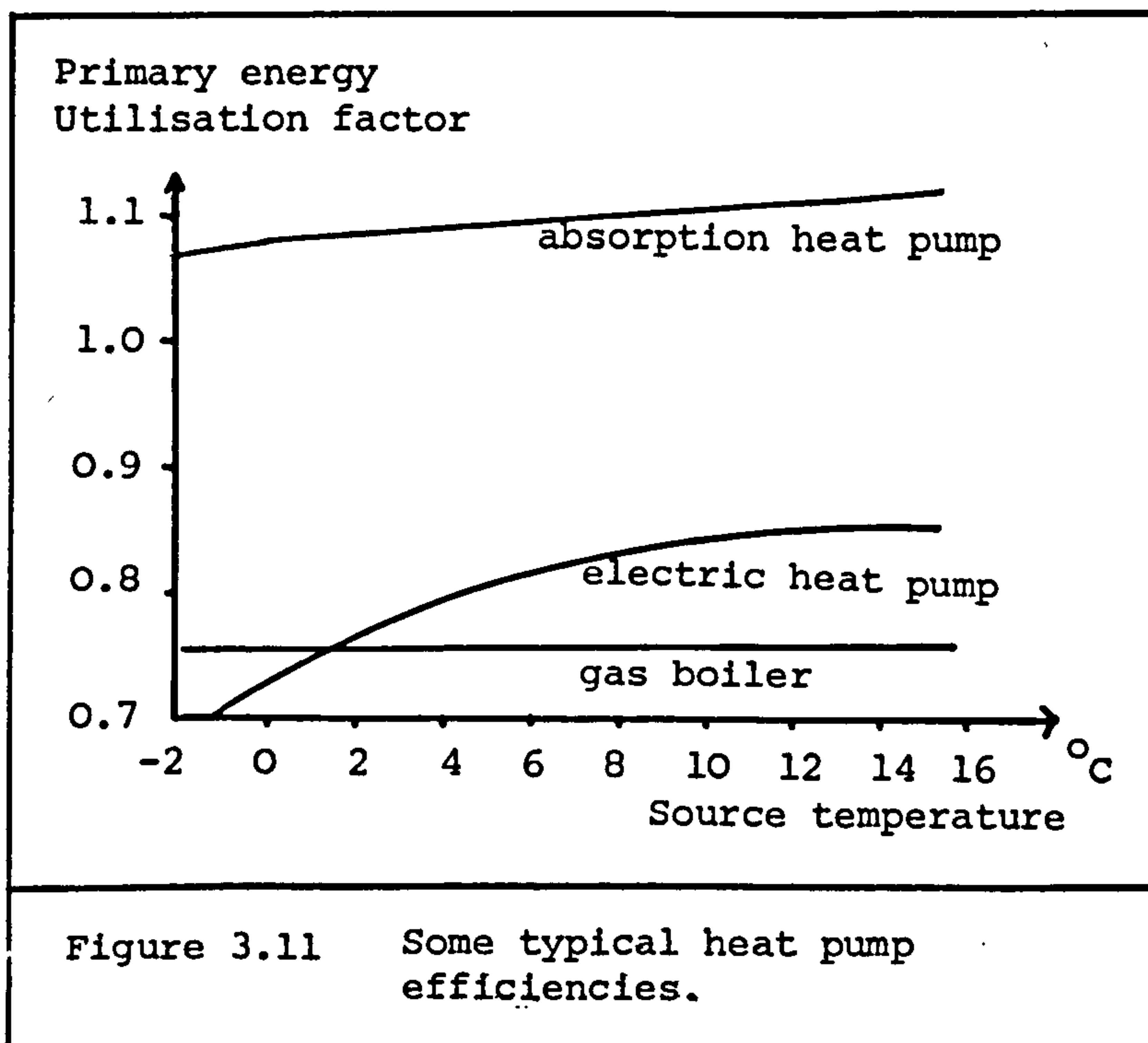
T_g is the generator temperature.

Thus since the term in brackets must always be less than one, the COP_H is always less than the COP_H of the vapour compression cycle

$$\frac{T_2}{(T_2 - T_1)}$$

However, in primary energy terms, an absorption cycle heat pump would still compare very favourably with an electric heat pump due, to the inefficiency of electricity generation in the UK. Typical efficiencies are shown in Figure 3.11

[12].



Various fluids have been used or proposed as suitable solvent/refrigerant pairs, the most common two of which are water/ammonia and lithium bromide/water. Ammonia has the disadvantage that it is toxic, whilst in the second pair, since water is the refrigerant, it cannot be used for evaporating temperatures below 0°C. Many other combinations have been considered, but there is undoubtedly scope for

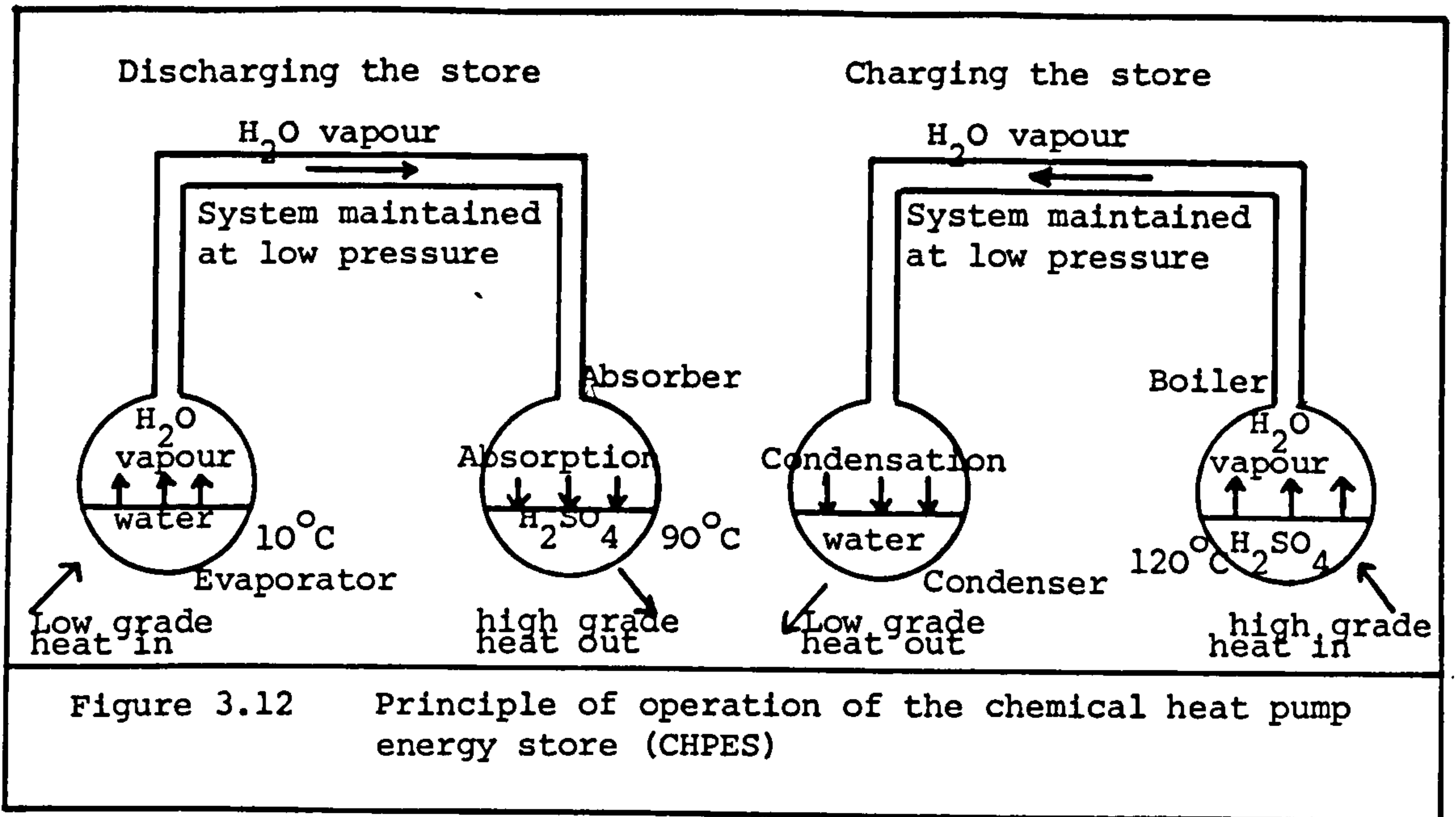
further research into other non flammable, non-toxic and non corrosive combinations having lower capital cost and wider operating ranges than those currently applied. Allied Chemicals in the U.S. are in the process of developing an organic absorber with a fluorocarbon refrigerant, [13, 14, 15], and solid absorbents [16] have also been proposed.

Currently, the maximum achievable temperatures are 90°C with water/ammonia and 120°C with lithium bromide/water [17].

Improvements in performance may be achieved by incorporating a two stage regenerator [18] (excess heat from the first stage of the regeneration process used to provide the heat input to the second stage) or multi stage regenerators, but it is thought that the key to future success lies mainly with the properties of the available working fluids.

Because an absorption cycle heat pump only requires a source of heat and no mechanical compressor, it has the advantage of being quiet and also of being relatively maintenance free. Also, because of the low COP, the heat output would be less dependant on the heat source temperature over a large part of the operating range. The evaporator could therefore be smaller and this should lead to a reduction in size and cost. Thus if small absorption cycle heat pumps can be produced cheaply enough they could become extremely competitive in the domestic market and for this reason a great deal of active research is being carried out both in the U.K. and in the U.S. by The British Gas Corporation and the American Gas Association respectively, both anxious to promote the use of such machines in their respective countries.

Another project making use of the absorption cycle was also under development at the Open University, but has since moved to the Cranfield Institute of Technology. The project, a joint venture with the Rutherford laboratories, was concerned with the development of a chemical heat pump/energy storage system, based on the dilution and reconcentration of sulphuric acid with water. [19,20] Whenever sulphuric acid is mixed with water heat is evolved, however this potentially cheap energy store suffers from a low storage density and if the acid is reconcentrated by distillation the overall efficiency also is low. However, by using it as a heat pump this sulphuric acid/water store can be greatly improved. The acid is mixed with water vapour and the source of heat for evaporation then becomes the environment. The reaction of the water vapour with the acid releases the chemical heat of mixing, plus the latent heat of evaporation of the water, and this total heat output is available at a temperature higher than the environment. To complete the cycle a source of high grade energy is required to reconcentrate the acid, diluted by absorption of water, and this represents charging of the store. The principle of operation is illustrated in Fig 3.12. overleaf with typical operating temperatures shown.



3.2. Prime Movers

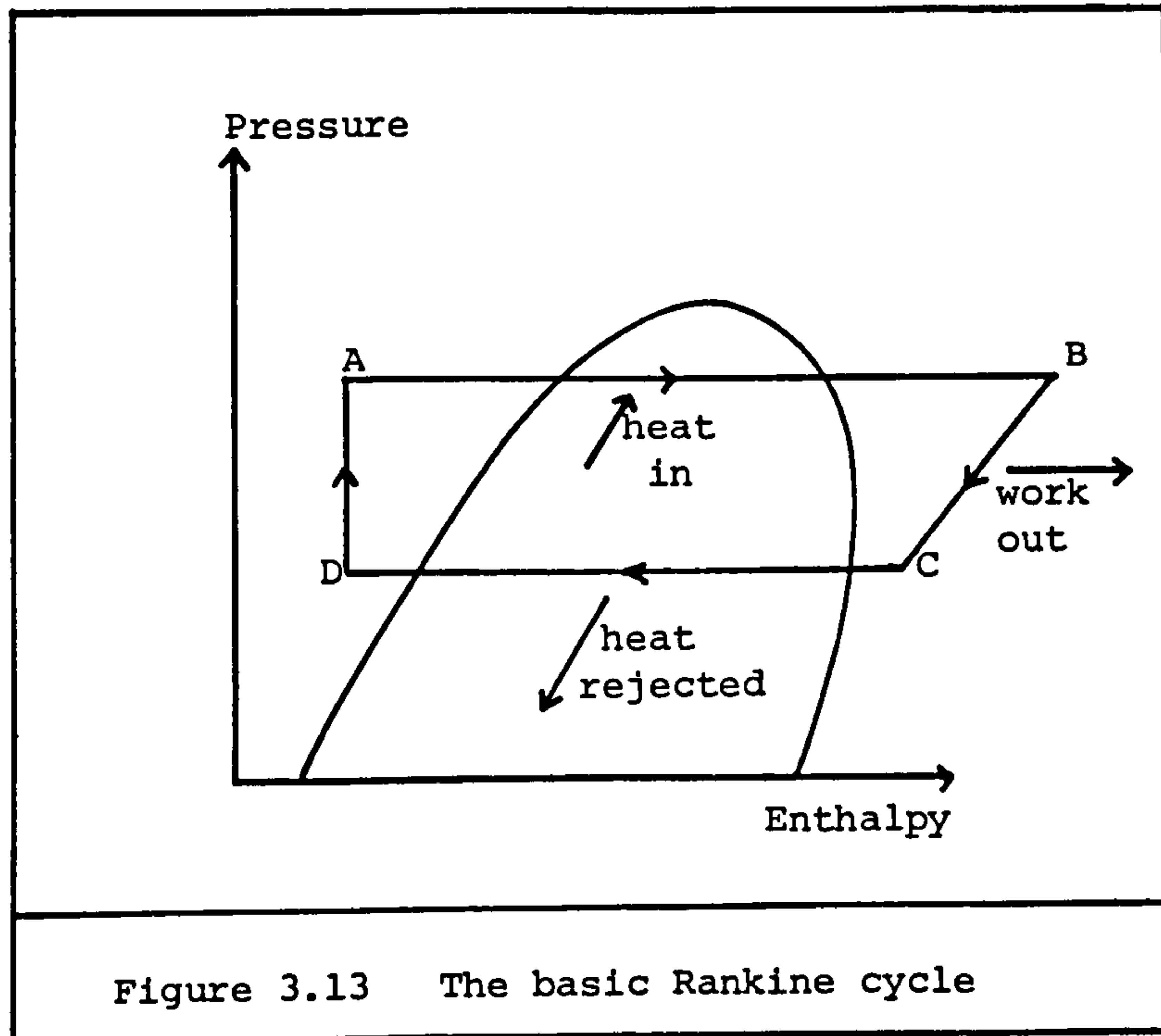
Most existing heat pump systems use electric motors as their drives. Electric motors have the advantage of being clean, quiet, cheap and reliable and can be used on all sizes and types of heat pump. One of the problems associated with them, however, is the lack of variable speed control to match a varying demand, which is expensive to achieve. By far the biggest disadvantage is that they are inefficient in terms of primary energy, since electricity generation in the U.K has a net thermal efficiency of around only 33% for the 20 best plants [21]. Thinking in terms of primary energy, fossil fuel drives are much more attractive for heat pump application, particularly if waste heat can be recovered from the engine's exhaust gases or cooling water. There are two categories of fossil fuel drives - internal combustion engines [22] (Otto, diesel) or external combustion engines (Stirling, Brayton, Rankine, Ericsson).

Otto (spark ignition) engines have the advantage of using tested, well known technology and are ideal for powering large (100 kW+) heat pumps. Indeed there are over 50 such machines already in operation in Germany. [23,24] Engine efficiencies of 30% can be obtained and noise and maintenance problems are minimal. For domestic applications however, the engine efficiency drops to around 20% and noise and maintenance problems prove difficult to solve.

The diesel engine is more efficient than the Otto cycle engine, however the same disadvantages apply and there is also a pollution problem, particularly for domestic applications. Thus it seems that internal combustion engines have a great potential for good energy savings in terms of large scale heat pump development. Mass produced car engines like that of the Ford 2274E [25] lend themselves particularly well to conversions to run on natural gas and are cheap, reliable and efficient. Smaller engines, like that described later in Chapters 5 and 6 are less efficient and more development work is required in this area.

The Brayton cycle engine (gas turbine) is used mainly for aircraft propulsion and power generation and as yet very little work has been carried out on heat pump applications. The American Gas Association [26, 27] are funding research on the development of a 35 kW, Rankine cycle heat pump, driven by a gas turbine. The engine was found to have an efficiency of 27.5%. Testing is still being carried out.

Rankine cycle engines are well suited to heat pump applications but are relatively underdeveloped as yet. Rankine cycles have had most application in the traditional steam engine and in steam generating sets. The basic cycle is well known and is illustrated in Figure 3.13.



The working fluid is evaporated at high pressure in a boiler, and may be superheated. It is then expanded, doing work. The low pressure gas is condensed, rejecting heat, and the resulting liquid is pumped back up to boiler pressure to repeat the cycle. The feed pump requires very little work input compared to the work output and the cycle can be further improved by reheating or employing a regenerative cycle. As with other external combustion engines a variety of fuels can be used and waste heat recovery is achieved directly from the condenser which is much cheaper than through the exhaust heat exchanger of a Brayton or other

cycle engine. The engine is quiet running and an efficiency of up to 25% should be attainable. Also being in a sealed unit, very little maintenance is required.

There are three main expander types - turbomachines, rotary valve devices and reciprocators. Of the first, three companies are so far using turbo-expanders, Ormat in Israel, Thermo-electron in the U.S., and Glynwed in the U.K. [28]. GEC are using a rotary vane expander for a solar cooling project, funded by NASA; and reciprocating expanders have so far only been used in two automotive engine projects. There are also a vast number of possible working fluids available and success or failure is mainly dependent on using the best configuration of expander, compressor and working fluid for the particular application.

The Stirling engine concept is well known and has undergone development by Philips and other companies for nearly 30 years. [29]. Laboratory tests on Stirling engines [30] indicate efficiencies of over 40%, but in practice efficiencies in the range of 20-35% are attained. They have good part load efficiency and, being external combustion engines, have low noise, little maintenance and good fuel versatility. [31,32]. Philips have used one of their existing range of Stirling engines (shaft power 5 kW) to drive a heat pump heating a pair of semi-detached houses in the Netherlands [33]. The heat pump is a water to water machine and with a fuel energy input of 20 kWh, the heat pump system is able to provide 28 kWh of heating. The engine

efficiency is about 25%. In another application in America, a free-piston Stirling engine is directly linked to an inertia compressor to provide a single compressor drive unit for a 10 kW heat pump. The engine efficiency in this arrangement is increased to 30-35%.

Ericsson engines have similar characteristics to Stirling engines, apart from having the regeneration process taking place at constant pressure and both suffer with the disadvantage of using helium as their working fluid, which is neither plentiful nor cheap.

There are many different possible combinations of power and heat pump cycles apart from those already mentioned and this has been an area of increased research activity over the past few years. [5, 11, 13, 34 and 35].

3.3 Heat Sources

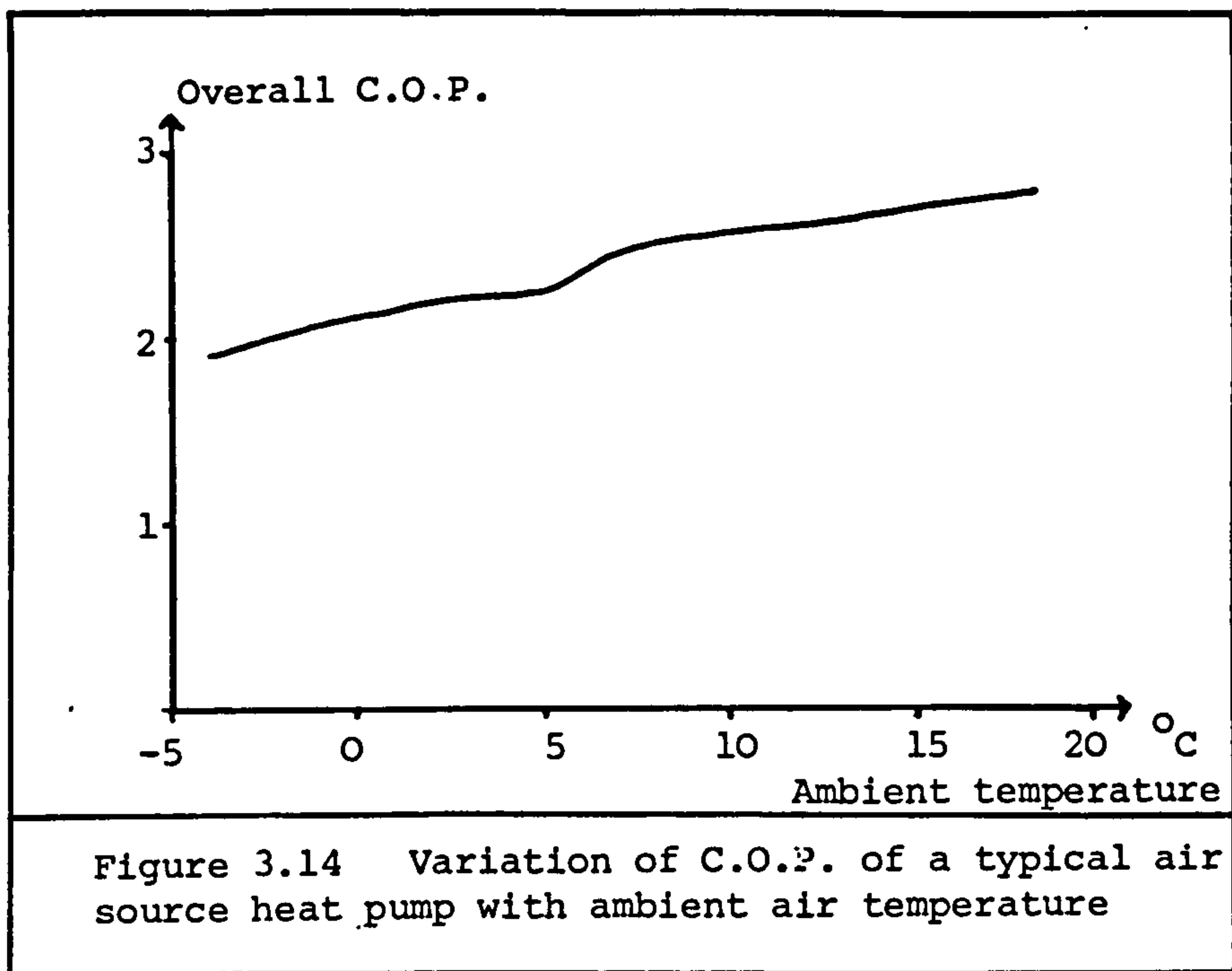
When choosing or designing a heat pump it should be remembered that ultimately it will form part of a larger system which includes a suitable source of low grade heat and a distribution medium. The choice of a suitable source will have many governing factors such as geographical location, climate, availability, application etc., however the most desirable features to aim for are:

a) The source temperature should be as high as possible, since a low temperature difference across the heat pump will give a good C.O.P.

- b) The source should contain a sufficient quantity of heat such that the extraction of the required amount will not significantly decrease the temperature of the source.
- c) The use of the source should not adversely affect the environment and should be freely available.
- d) The temperature of the source should be reasonably constant since the heat pump will operate most efficiently over a narrow range of temperatures.

The most commonly used natural sources are air, water, (either as flowing streams and rivers or as ground water), or the soil. Other possibilities include industrial waste heat recovery from exhaust air, or effluents.

Air is a universal, cheap and abundant source and as such is used by most commercially available heat pumps at the present time. However it does have its disadvantages. Because air has a low density and therefore a low heat capacity, a large volume is needed. Also, air is subject to wide and rapid temperature fluctuations, and, as the outside air temperature drops, so the demand for heat inside a building normally rises. Thus the source and sink temperatures are generally out of phase, and, since the C.O.P. depends upon the temperature difference between the evaporator and condenser;- as the ambient temperature drops, the temperature difference increases and the C.O.P. deteriorates. This is illustrated in Figure 3.14 overleaf. [36]



There are however various ways of overcoming this problem (depending on the particular application) by using waste air from buildings, solar preheating or by using a form of supplementary heating at times of low ambient temperatures. Appendix II gives some idea of the estimated number of hours during a heating season that a heat pump would have to run at various ambient temperatures and Appendix III shows the average hourly temperatures for each month of the year, computed over a 25 year timespan from Meteorological office data (Heathrow weather station).

The other major drawback of using ambient air as a heat pump source relates to the fact that the U.K. suffers a predominantly damp atmosphere with a relatively high humidity. Thus at low ambient temperatures, if the air is cooled below its dewpoint, the moisture contained in the air condenses out and may freeze on the heat exchanger forming an insulating layer of ice and effectively preventing further

heat transfer. To counteract this the air flow over the heat exchanger is usually designed to be as large as possible thereby minimising the temperature drop. However, if ice does form, in sufficient quantity to poses a problem, defrost can be achieved by adopting any of the following methods:

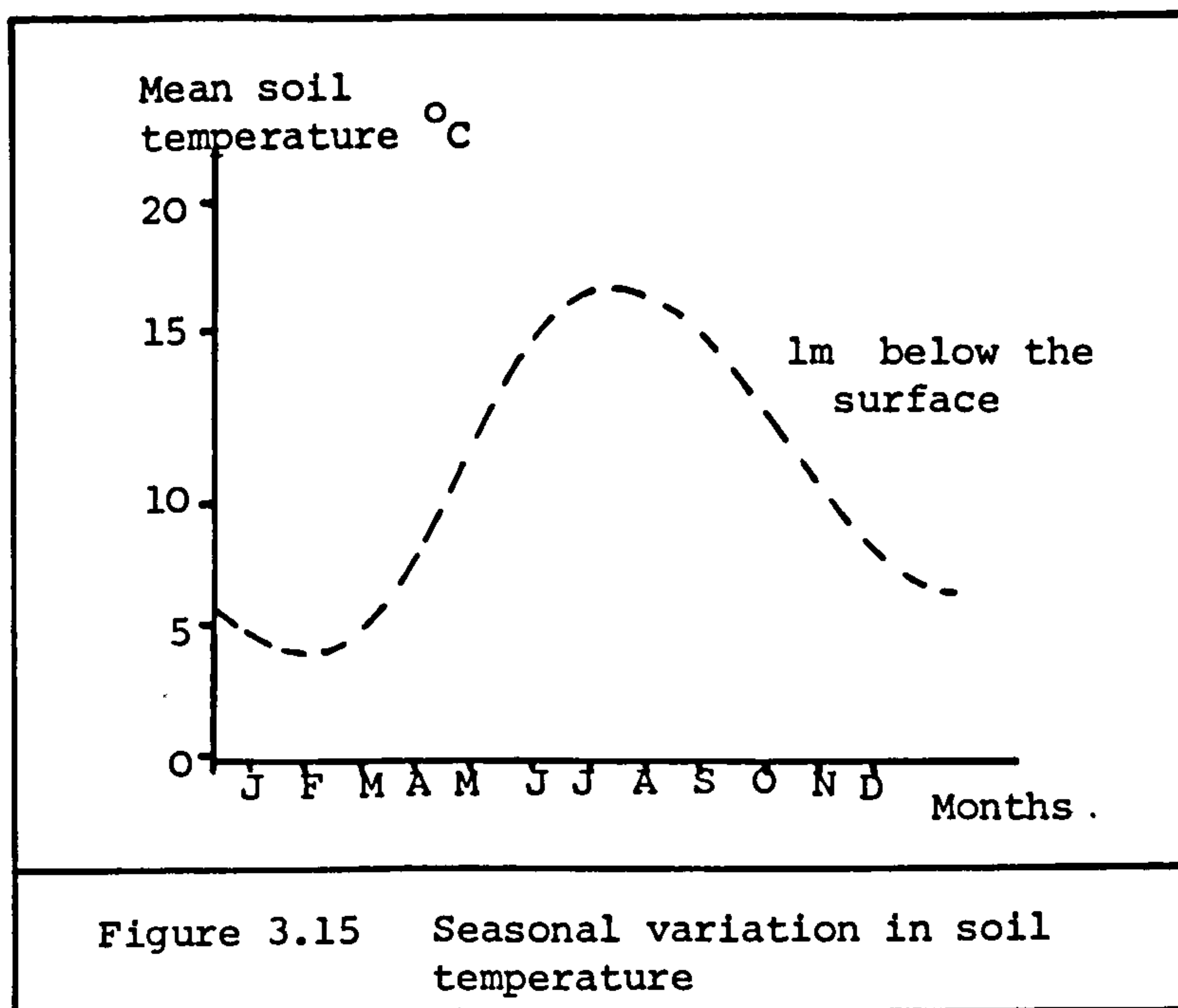
- a) by reversing the heat pump, thereby heating the evaporator.
- b) by incorporating electric heating coils in the evaporator design.
- c) by passing hot gas from the compressor directly to the evaporator.
- d) by mechanical scraping of the evaporator fins [37].

The rate of frequency of defrost depends upon the design of the evaporator and the evaporator conditions. Ambrose gives some typical figures for this, [38]. However there is still a great deal of debate on this topic. Trials by the Electricity Council [39] concluded that compressor power during defrost accounted for 2% of the total heating power.

Water, with its high specific heat capacity is an excellent heat pump source, but its major drawback is that of availability. A flowing river or stream will generally have a temperature in the range 5-10°C but unfortunately its application would be limited to buildings sited close to the source. Also, care would have to be taken to ensure that the natural ecology of the river/stream was not disturbed, and that the rate of flow was adequate to prevent freezing.

Static water such as a lake would be more prone to freezing than flowing water and extra pumping would have to be employed to prevent this. Mains water would be an extremely good source for a heat pump since its temperature very rarely reaches freezing point however the cost may be prohibitive since it would depend on agreement with local water authorities. This would also be true of using ground water and here the pumping costs would again be an added expenditure. Nevertheless, this is proving to be a popular source in Denmark and West Germany, [40]. As previously mentioned, heat from sewers has also been utilised as in the Nuffield College heat pump at Oxford [41]

There have been many heat pump installations using the ground as a source of heat since it has the advantage that the temperature of the soil remains relatively stable and is unlikely to drop below 5°C. Also the soil temperature lags behind the air temperature by about two months, thus the minimum soil temperature occurs well past the peak of the heating season. This is illustrated in Figure 3.15 overleaf which shows soil temperatures measured in Belgium. [42]



The heat transfer characteristics of soil depend upon its ground water content. A dry soil has a lower conductivity than a wet soil and longer piping would be needed to achieve the same heat transfer, thus areas with high water tables would be most appropriate. The source can be utilised by burying metal or plastic pipes in a horizontal network at depths ranging from 0.5 to 2 metres and then circulating the refrigerant or some other heat collecting solution (e.g. brine or ethylene glycol) through them. There have been many studies and computer models to determine the best depth, pipe size layout etc.[43-46].

The problems normally associated with air source heat pumps, namely noisy evaporator fans and defrosting are eliminated with ground source systems but the major disadvantages are that the systems are inaccessible for maintenance and also installation costs are high since a large collection area is generally required. Thus lack of space can also be

prohibitive for many applications. A vertical coil would therefore have obvious space and economic advantages and research work at Queen's University, Belfast [47] on vertical 'U' tube coils suggest that a reduction in ground coil surface area of between 10 and 20 can be achieved for the same heat uptake.

The first ground source heat pump to operate in the U.K. was that built by Sumner [48] in 1954 to heat his bungalow in Norwich. An antifreeze solution of methanol was circulated in 450 feet of copper pipe buried 3 feet in his garden and he was able to achieve an average C.O.P. of 2.8. Another ground source collector was used to heat a laboratory belonging to the Electrical Research Association (ERA) at Shinfield in 1951 [49]. Here 500 feet of pipe was buried horizontally 3 feet below a lawn, the pipes being 1 foot apart. Again a good C.O.P. was achieved ranging from 2.2 to 3.8. No adverse effects were noticed on the soil due to the lowering of the temperature, although it was reported that the lawn appeared slightly convex at the end of the heating season!

There are a great many cases where industrial waste heat can be put to good use by the use of heat pumps and many of these are dealt with in more detail in Chapter 4. It may be possible to utilize waste heat from an industrial process at some point in the process, or to provide heating in another area of the building. Examples of these are waste heat from freezer rooms, from freezing ice rinks, exhaust ventilation air etc. One particularly good source which has received a

lot of attention is power station effluent. If the heat rejected is at a high enough temperature it may be used directly for district heating schemes, but if the temperature is not high enough a heat pump can be used to upgrade it.

[50-53]

The use of heat pumps with solar collectors has also been a considerable area of interest. [54-57] For example, the utilisation of solar energy to preheat the air for an air source heat pump, or to directly heat the water for a water source heat pump, would increase the performance of the collectors, but the cost of such a system would probably prove prohibitive at the present time. [58]

3.4. Working Fluids

The life blood or working fluid of any heat pump is known as the refrigerant. By evaporating it, heat is extracted from the heat source and by condensing it, heat is released where it is required. There is a wide choice of refrigerant suitable for heat pump applications, and in making such a choice the following properties are most important:

- (a) The refrigerant must be capable of liquifaction at a condenser temperature of the gas.
- (b) It must have no corrosive effect on the pipework, valves, seals etc.
- (c) It should have a high latent heat.
- (d) It should be non-toxic and non-inflammable.

- (e) It must not solidify during any stage of the process.

Refrigerants are classified by number according to the Ashrae Standard List of Refrigerants, but are more commonly known by their trade names e.g. 'Freon 12' (Du Pont), 'Arcton 12' (ICI) or 'Genetron 12' (Allied Chemicals). The number is derived from the chemical formula. The first digit refers to the number of carbon atoms minus one, the second digit is the number of hydrogen atoms plus one, and the third digit is the number of fluorine atoms. If the first digit happens to be zero, it is omitted. eg. CCLF_2CF_3 becomes refrigerant number $(2-1)(0+1)(5) = 115$. Azeotropic mixtures are represented by numbers 500 - 504 and the inorganic refrigerants are represented by the 700s', by adding 700 to their molecular weight. Ammonia for instance, is R717.

The properties of a refrigerant can be shown plotted on a Mollier diagram, a schematic of which is shown in Figure 3.16 overleaf.[59] Charts of this nature plus tables can generally be easily obtained from all the refrigerant manufacturers. Table 3.1 also overleaf sets out some of the properties of the most commonly used refrigerants.

The halogenated hydrocarbons, or halocarbons (eg. R12, R22, R502) are probably the refrigerants most widely used today. They were first used in the 1940's, to replace the more toxic inorganic refrigerants such as ammonia or sulphur dioxide which were popular then. Nevertheless, in spite of its

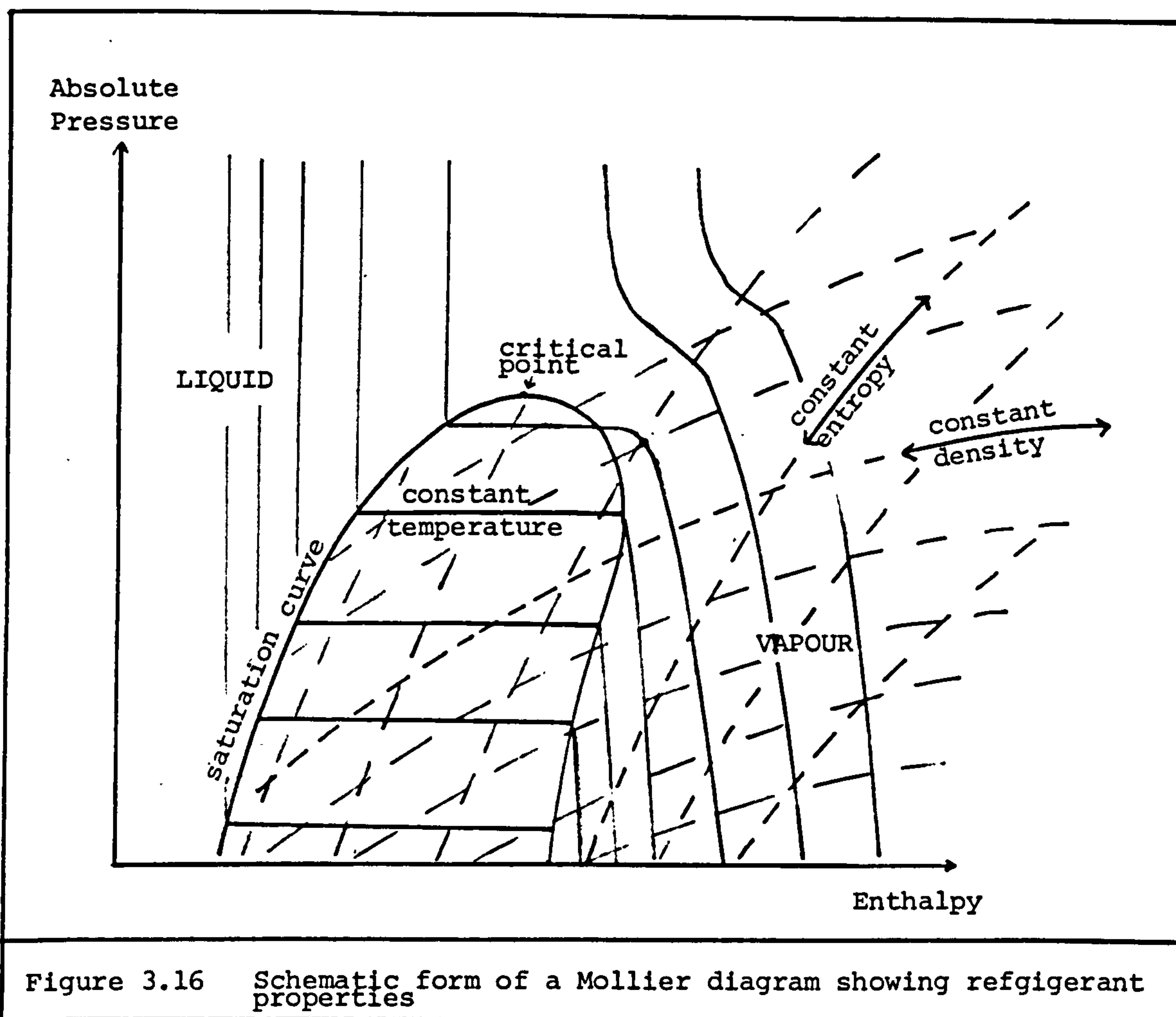


Figure 3.16 Schematic form of a Mollier diagram showing refrigerant properties

Table 3.1 Table of Comparative Refrigerant Properties (36)

(-13°C evaporating temperature, 30°C condensing temperature)

	REFRIGERANTS				
	11	12	22	114	502
Chemical formula	CFC_3	CF_2Cl_2	CHF_2Cl	$\text{CF}_2\text{Cl}-\text{CF}_2\text{Cl}$	R22+R115
Chemical name	Trichloro-fluoro-methane	Dichloro-difluoro-methane	Chloro-difluoro-methane	Dichloro-tetrafluoro-ethane	48.8% R22 by weight
Critical temperature(°C)	198	112	96	146	82
Evaporator pressure (bar)	0.21	1.82	2.95	0.47	3.49
Condenser pressure (bar)	1.25	7.44	11.9	2.53	13.1
Compression Ratio	6.19	4.08	4.03	5.42	3.75
Refrigerating effect (KJ/kg)	155	116	163	100	106
COP	5.03	4.70	4.66	4.49	4.37

corrosive and highly inflammable nature, ammonia is still used in large industrial applications. Sulphur dioxide, even though it is non-flammable, was even more toxic than ammonia, forming sulphuric acid if dissolved with water. Although the halocarbons are therefore considerably safer than these earlier working fluids there is still concern at the detrimental effects they may have on the ozone layer which surrounds our planet [60], and this has resulted in a ban on the use of non-essential chlorofluorocarbon aerosols in the United States [61]. There are also fears that some of them may have carcinogenic properties.

R12 and R22 are the two most popular refrigerants for heat pump applications to date. R12 is widely used for refrigeration purposes and operates in the evaporating range - 35°C to +10°C. R22 operates at a higher pressure than R12 but suffers from high superheat temperatures. Nevertheless it has been used successfully for years in air source heat pumps. At higher condensing temperatures R11, R113 and R114 are the main contenders. [62] Non-azeotropic refrigerant mixtures have also been suggested for use in the future, operating on the Lorentz cycle, but would used larger, counter-flow heat exchangers [63].

3.5 Compressors

A further essential item for any heat pump system is the compressor, needed to pump the refrigerant around the system, and to increase its pressure. Any compressor which is

suitable for refrigeration purposes can be used for heat pump application but it must be remembered that heat pumps normally operate for longer hours, at higher condensing temperatures, and at greater pressure ratios than refrigerators. Therefore any compressor chosen must be capable of withstanding more stress. The four most commonly used types of compressor are: reciprocating, centrifugal, rotary sliding vane or rotary screw.

3.5.1 Reciprocating

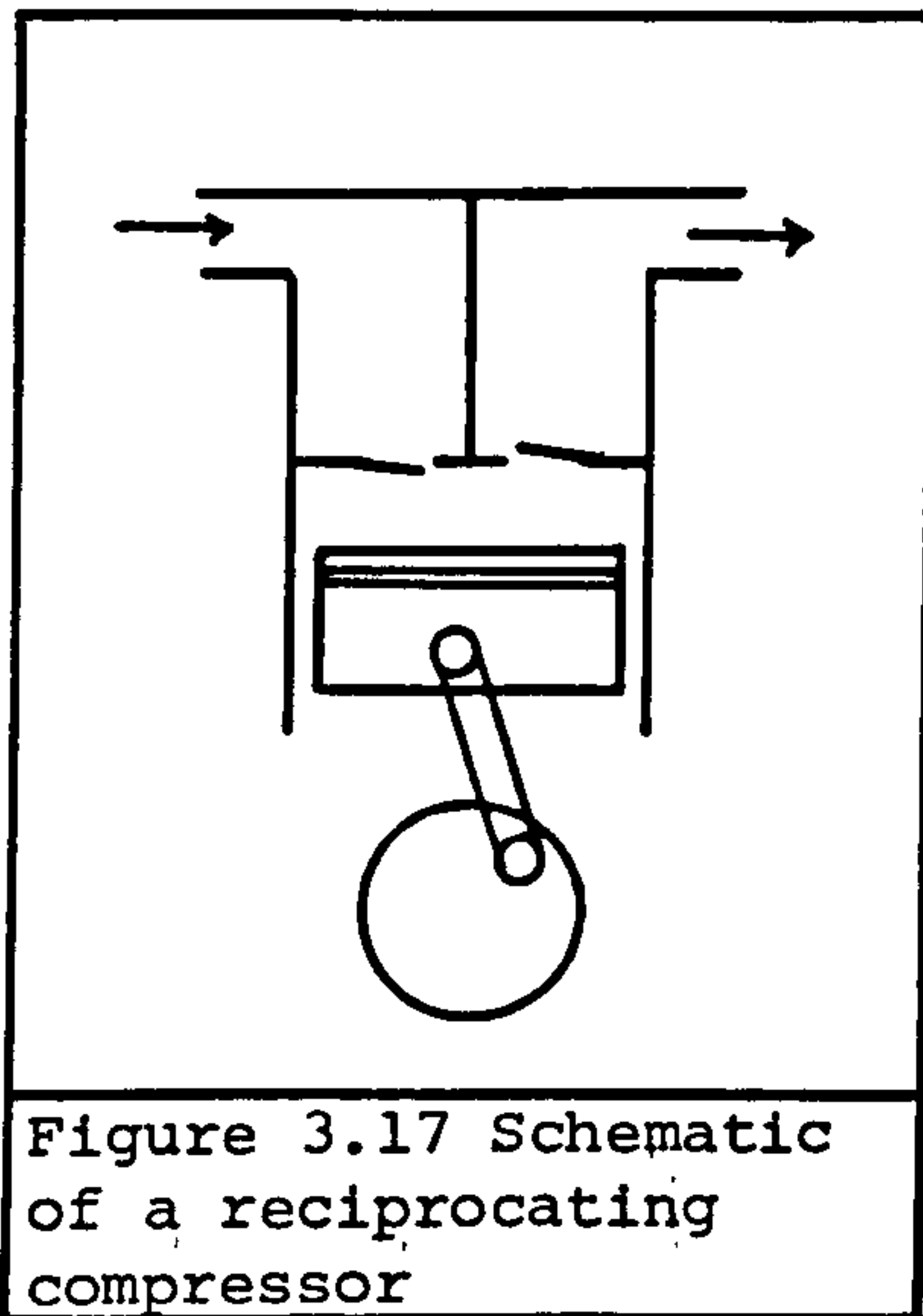


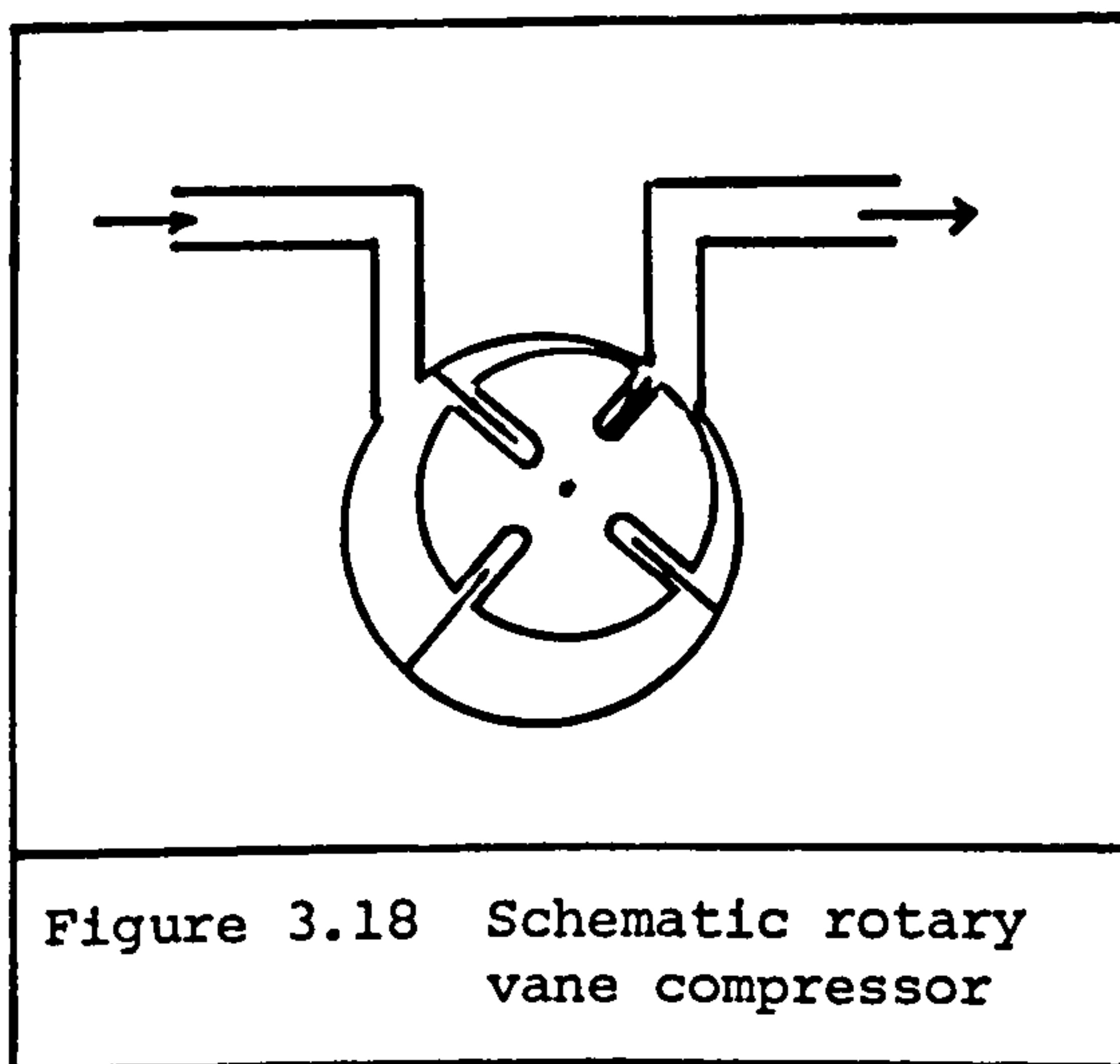
Figure 3.17 Schematic of a reciprocating compressor

This type of compressor comes in various sizes, from a few watts, up to 150 kW and since it is a positive displacement machine it can maintain a high discharge heat, and high condensing temperatures at low loads. There are three types of reciprocating compressors: hermetic, semi hermetic or open.

In the hermetic type, the compressor and motor are both contained inside a hermetically sealed outer case. The motor is cooled via radiation or convection or by suction cooling (drawing cold refrigerant over the motor). This type of compressor is quiet and reliable with overall efficiencies of 40- 60% and as such has found many applications in domestic refrigeration. The semi hermetic type is normally found in larger units, up to 150 kW, for commercial and industrial application. The compressor and motor are still within one

casing, but this is bolted, not welded. It is noisier and more expensive than the hermetic type, but disassembly is easier for maintenance and repair and overall efficiencies can be as high as 70%. Open reciprocating compressors have the crankshaft passing through shaft seals in the casing with a separate motor coupled directly or indirectly by belts. This type of compressor gives greater flexibility in allowing a wide range of prime movers to be used, however it is even more expensive than the others and, since it is not completely enclosed, is subject to leakage of the refrigerant through the shaft seals.

3.5.2. Rotary Vane Compressors

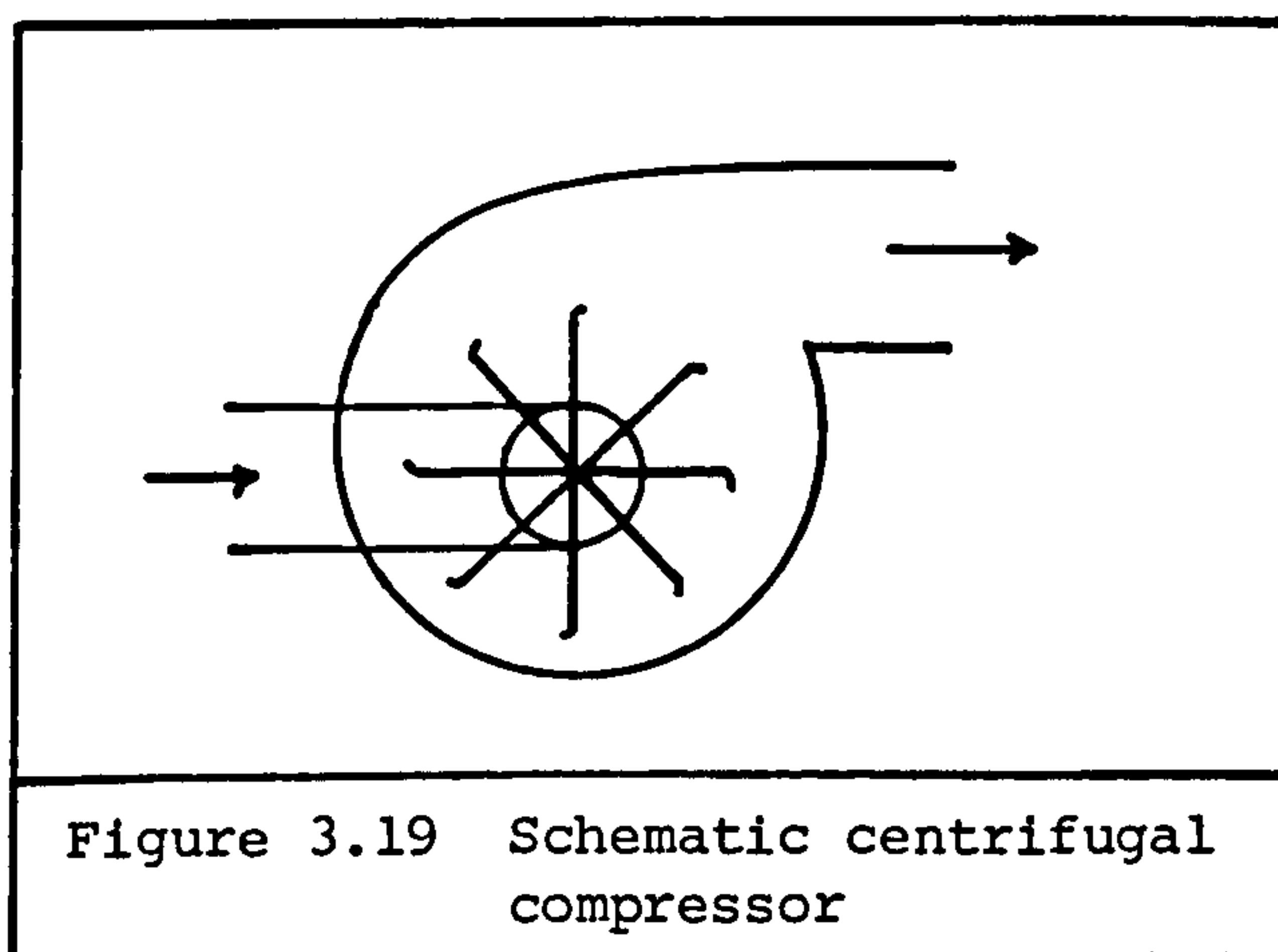


These are small compact compressors, also known as sliding vane compressors, which normally operate with low compression ratios and low pressures. They have a high volumetric efficiency, are well suited to high rotational speeds and are considerably more robust than reciprocating compressors.

Compression takes place between the sliding vanes which are commonly made from reinforced phenolic resin. The number of blades is a compromise between volumetric efficiency (increased number of blades) with friction losses. The blades are sealed to the cylinder by a thin film of oil on

the inside surface of the cylinder. Applications of under 5 kw are normally most suitable for this type of compressor although one recent development by Prestcold/Denco Miller extends this range to 10 kW. A particularly interesting use of a rotary vane compressor is in the Rovac system previously discussed in Section 3.1.2. Here a combined compressor/expander is used in an air cycle machine. [9]

3.5.3 Centrifugal Compressors



Centrifugal or turbine compressors are best suited to large throughputs where there are comparatively small temperature differences between input and output temperatures.

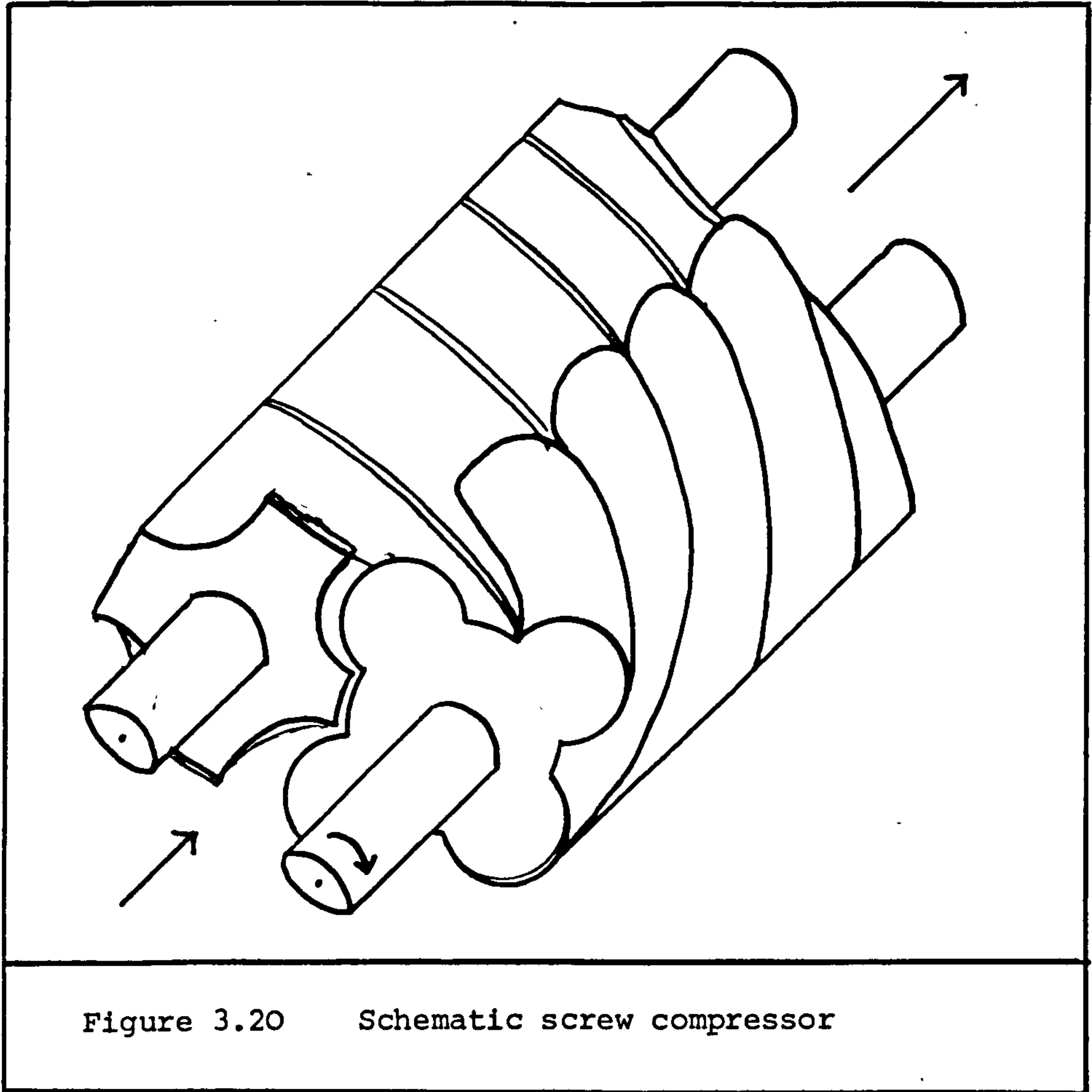
They are often used in place of positive displacement compressors for very large capacities in the 300 kW - 20 MW range.

Single-stage or two-stage compressors are available and multi-stage compressors can be made by linking stages with external pipework or incorporating the compression stages within the same housing with interstage gas injection between the rotors.

3.5.4 Screw Compressors

Screw compressors consist of two vertical rotors, male and female, running together inside a sealed sleeve. The male

rotor has on it a number of lobes and the female is fluted



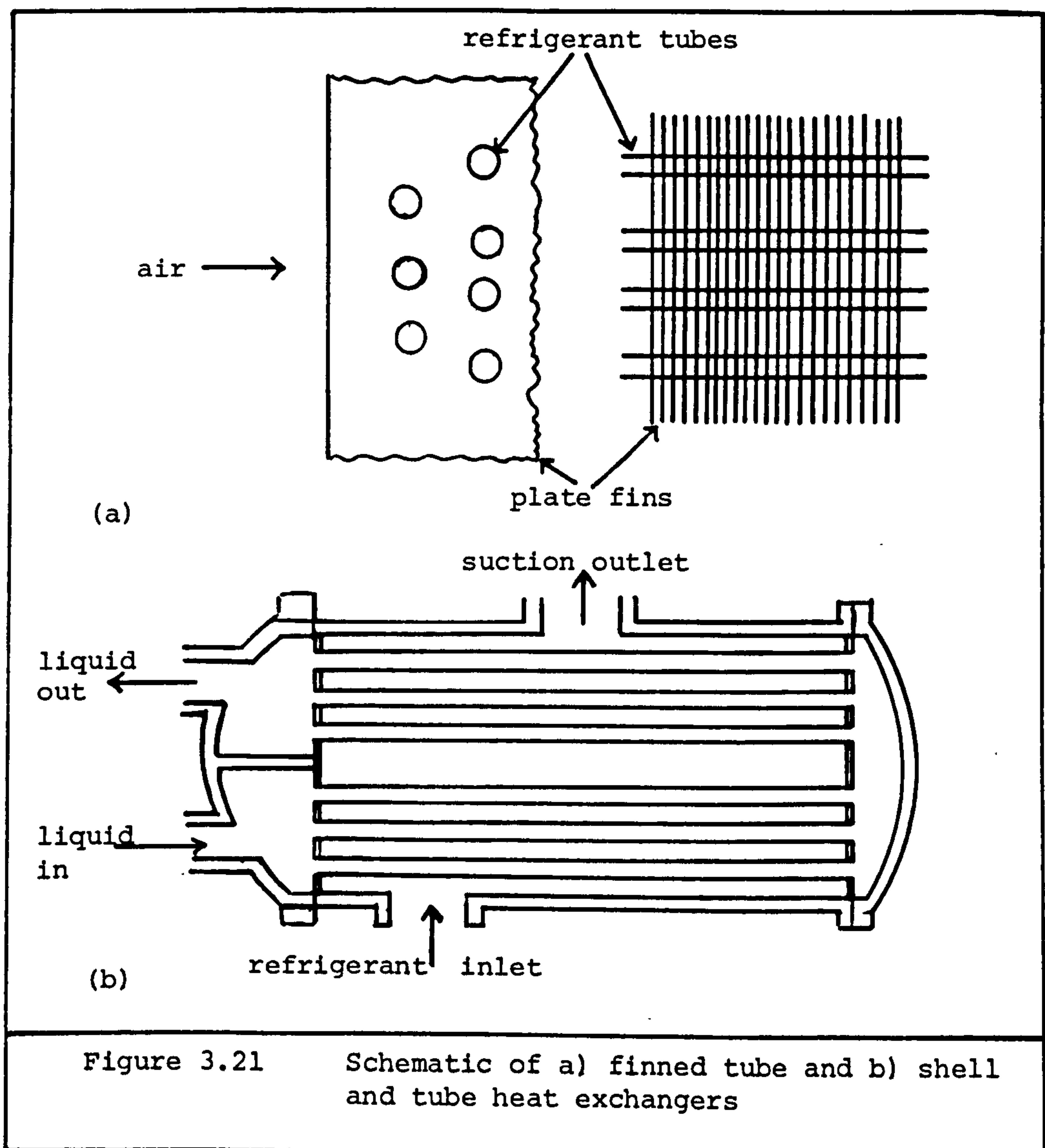
and compression occurs in one channel by interference between the two rotors. Oil injected screw compressors (the oil, continuously pumped in, forms a seal between the rotors and directly removes the heat of compression from the refrigerant) are ideal for medium and large heat pump applications (300 kW - 2500 kW range) but are extremely expensive and noisy.

3.6 Other Components

3.6.1 Heat Exchangers

Heat exchangers are devices which permit two 'fluids' to transfer heat from one to the other, without coming into

direct contact. There are usually two heat exchangers in any heat pump - a low temperature heat exchanger (evaporator) and a high temperature heat exchanger (condenser). There are two main types in common use - finned tube heat exchangers are used for heat transfer between air and a refrigerant, and the shell and tube type are used for liquid-refrigerant heat transfer.



There are a number of variations possible in the fin type heat exchanger since the finning can run across all the coils (as in Figure 3.21(a)) or the fins can be wrapped around each tube. The extensive use of fins is to increase the surface

area of the heat exchanger in order to compensate for the poor heat transfer coefficient of air. A fan is generally used to force the unwanted air to circulate over the coils, however this can prove to be a source of noise. To overcome this, Blundell has considered a flat plate collector using natural convection. [64] But the increased cost of such a system, does not prove to be economic at present.

The shell and tube heat exchanger shown in Figure 3.21(b) is by far the most common type used as condensers. The refrigerant is enclosed in a shell through which tubes containing the other liquid pass. A variation on this is the tube in tube heat exchanger. This then becomes a counterflow heat exchanger consisting of two concentric straight tubes.

When choosing the appropriate heat exchanger there are a number of factors to be considered, most importantly their heat transfer characteristics. This depends on the geometry of the heat exchanger, operating conditions, materials etc. All simple heat transfer processes can be described by the basic heat transfer equation

$$q = U A \Delta T$$

where q is the heat flow, A the effective heat transfer area, and ΔT the average effective temperature difference. U is a heat transfer coefficient, a complex combination of many factors including flow rate, temperature, materials.

Equations for determining U and heat transfer equations for a number of heat exchangers are given in standard text books.

[65,66] To increase the amount of heat transfer, there are a

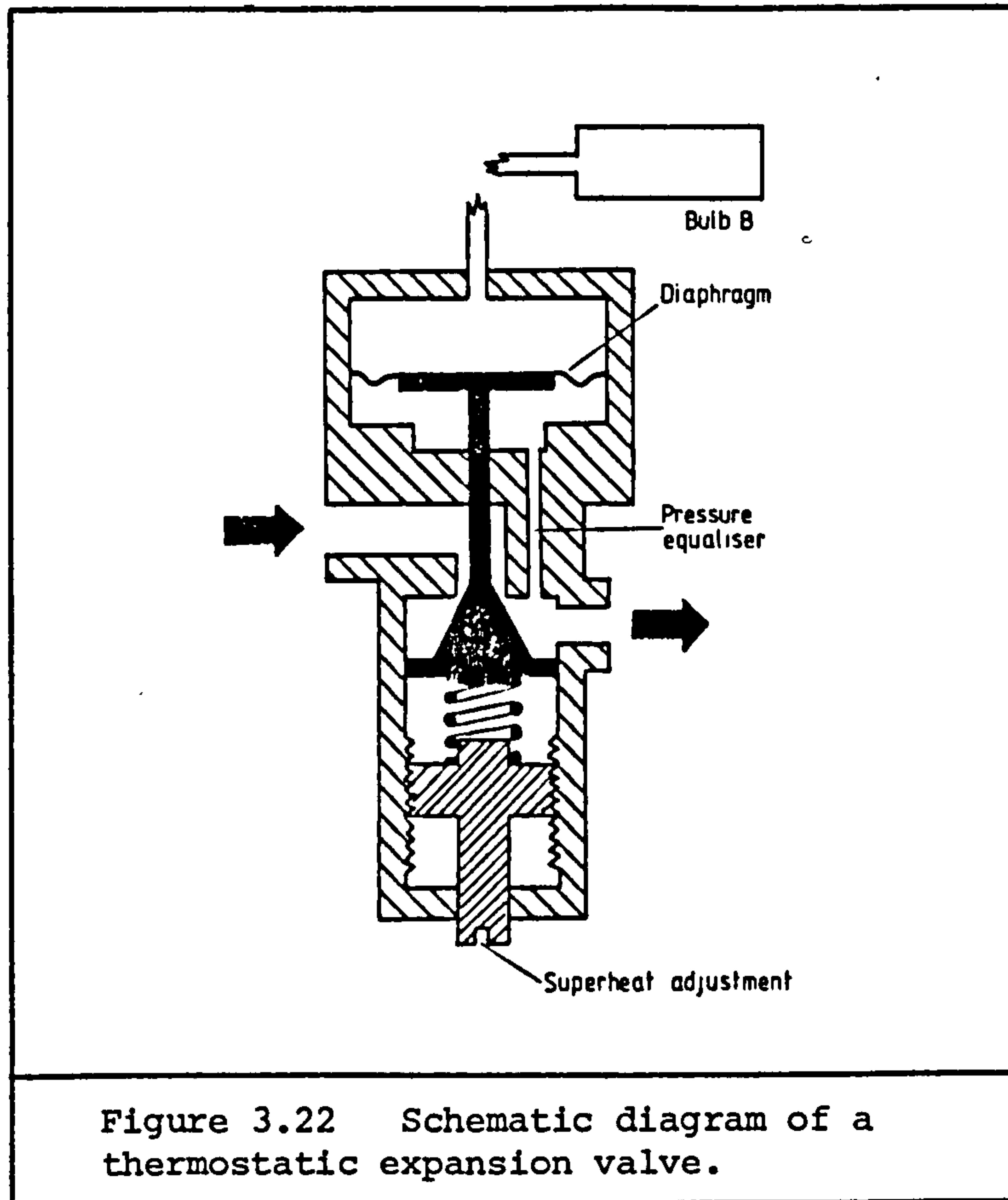
number of options open to the designer of a heat exchanger which are immediately apparent from the equation e.g. by increasing A , the surface area, or ΔT the temperature difference.

When considering the materials used great care must be taken to ensure that no corrosive reactions can occur between the material and the refrigerant e.g. copper is the most widely used material due to its high thermal conductivity, but it is totally incompatible with ammonia. Once the type of heat exchanger has been fixed it must be correctly sized to ensure that the final product is an economic heat exchanger which will meet the required performance characteristics. By increasing the size of the heat exchanger, i.e. increasing the surface area, and reducing the difference between evaporating and condensing temperatures, a high C.O.P. will be achieved, but at increased cost. Several optimisation exercises have been published [67,68], and an interesting piece of information from Blundell's work was that the evaporator on American air to air heating - only heat pumps should be almost doubled in size for U.K. conditions.

3.6.2. Expansion Valves

In order for the refrigerant to be able to pick up heat from the low temperature source in the evaporator, its pressure and temperature must be reduced and this is achieved by the inclusion of an expansion valve in the system. For most heat pump applications a thermostatic expansion valve (TEV) such as that shown in Figure 3.22 is the most commonly used, which

not only reduces the pressure and temperature of the



refrigerant, but also regulates the flow so that the evaporator receives just the required amount of refrigerant that can be evaporated. The TEV consists of an orifice which is restricted by a needle or plunger, a diaphragm and a sensing bulb, connected by a capillary tube to the diaphragm. The bulb senses the temperature of the vapour leaving the evaporator and is usually filled with a liquid/vapour mixture. An increase in superheat causes increased pressure in the bulb which depresses the diaphragm, thus opening the needle valve and hence increases the refrigerant flow. If there is a decrease in superheat the reverse happens.

3.6.3. Miscellaneous Items

To connect up all the various components already considered

pipework must be used which will not leak any refrigerant and will not corrode or cause corrosion. Excessive pressure drops must be avoided by using large enough pipe diameters and minimising the number of sharp bends in the system. Also integrated into the pipework may be extra components not already mentioned such as oil separators, valves, filter dryers, flowmeters, and sight glasses (to allow for visual inspection for unwanted vapour bubbles ie. to ensure 'solid' liquid exists). Generally copper tubing is chosen but particular care must be given to joints which should be silver soldered or brazed to withstand higher pressures.

To increase circulation rates of air or liquid, fans or pumps may also be included in the system, but their energy use and noise should be minimised. For most air source heat pumps some sort of fan will generally have to be used due to the low heat capacity of air. The choice of fan will depend on available space, flowrate and pressure drop, and may be centrifugal, propellor, or axial. Large, slow running fans are quieter and more efficient than smaller units providing the same air flow, but their choice will depend on cost and space limitations. Also it must be borne in mind that fans and pumps are particularly prone to accidents, hence they should be sited so as to allow ease of service and maintenance.

CHAPTER 4 THE MARKET FOR HEAT PUMPS

4.1 Applications and Potential Savings

The over-riding objective for any heat pump research is that it will lead to cost effective appliances that are also marketable commodities. It is only by achieving this that large scale application will be possible, allowing heat pumps to make a significant contribution to national energy saving.

As already mentioned most existing heat pumps in this country are electrically driven, and whilst providing an efficient use of electricity, do not necessarily provide the most efficient use of primary energy as can be seen in Table 4.1.

Table 4.1 Primary Energy Efficiencies of Different Heating Systems

Fuel	Primary Energy Unit	Delivered Energy	Heating System	Useful Energy	Cost/MJ (useful)	
					Dom. P	Comm. P
Gas	1.0	0.9	Gas Boiler (effic: 75%)	0.68	0.36	0.34
Elec	1.0	0.25	Electric H.P COP = 3.0	0.75	0.39	0.28
Gas	1.0	0.9	Gas H.P COP=1.2	1.08	0.23	0.21

(1981 prices)

Thus it is evident that heat pumps offer a great potential for energy saving, in terms of space and water heating, particularly in the domestic and commercial sectors. The domestic sector of the UK consumes some 30% of primary energy, 85% of which is used for space and water heating and in the commercial sector 75% of its 11% of primary energy is

used for the same purpose [1]. Similarly in the industrial sector, which consumes some 40% of primary energy, 15% of this is used for space and water heating. Approximately 60% of all fuels used in the UK are consumed to provide heat, and over half of this is used for space and water heating under 80°C. A detailed breakdown is given by Leach and is shown in Table 4.2 [1].

Table 4.2 Energy Breakdown by Fuels and End Uses - 1976

	Solid Fuels	Liquid Fuels	Gas	Elec.	Heat*	Total %
Low Temp. Heat (under 80°C)	8.2	10.5	12.0	3.1	1.0	34.8
High Temp. Heat (over 80°C)	7.4	7.5	6.5	1.1	2.5	25.0
Electricity (lighting etc.)	--	-	-	8.1	-	8.1
Transport	0.1	20.8	-	0.2	-	21.2
Other-chemical feedstocks etc.	--	9.5	1.5	-	-	11.0
TOTAL	15.7	48.3	20.0	12.5	3.5	100

* co-generation of heat and electricity in industry

With future developments, heat pump output temperatures up to 200°C should become feasible. So, if we consider the demand for heat in this temperature range we have, not only the space and water heating load in domestic, commercial and industrial premises, but also relatively low grade process heat in industry.

The energy used for these purposes has been estimated by Bush [2] and is shown in Table 4.3 overleaf. The overall total, 28 Gigatherms, represents 38% of the national primary fuel

consumption.

Table 4.3 U.K. Demand for Low Grade Heat (1975)

Market Sector	Heat used in 1975 (Gigatherms)	
	Heat supplied	Primary fuel
Space and water heating ;		
Domestic	12	18
Commercial	5	7
Industrial	3	3
Total	20	28
Process heat up to 200°C	3-4	3-4

From the Table it can be seen that the greatest potential for energy saving by heat pumps is in the domestic space and water heating sector which accounts for 56% of the total U.K. demand for low grade heat (equivalent to 18GJ). Indeed one study [3] estimated that a 7% saving could be made in the UK's energy budget by using heat pumps to provide all domestic heating (the same study estimated a 4% saving from improved thermal insulation in houses).

4.1.1 Domestic Sector

Heat pumps can only succeed commercially in the domestic sector by competing in the central heating market, which is dominated by "wet" combined space and water heating systems. Sales of such systems seem likely to plateau in the mid to late eighties by which time there will be a substantial replacement market for heat pumps to compete with. If a 10% share of the market were to be captured by heat pumps selling at say £300-£500[†] each, the annual turnover would be £20-30

[†] Prices and costs quoted throughout this chapter are based on 1981 figures.

million, which is not unattractive. There are a number of factors however which could effect this penetration of the market by gas engine driven heat pumps. Firstly they will be in direct competition with electric heat pumps, which are far more advanced and can be expected to improve on their current technical performance. Also they will have to compete with other fuel saving concepts such as increased insulation measures, (reducing fuel consumption and therefore giving less incentive to install heat pumps) solar panels, and combined heat and power schemes.

There are many criteria which a domestic gas fired heat pump will have to meet. It will have to be quiet, clean, of comparable size to a boiler package, and reliable (i.e. not require an excessive amount of maintenance). Since the market is dominated by "wet" systems, heat pumps must be compatible with these and it would be preferable (especially for the replacement market) if output temperatures similar to current wet systems could be attained (70-80°C).

However, it would be more economic in many cases to use lower radiator temperatures and improve the insulation of the house. To achieve the higher temperatures would reduce the efficiency of the heat pump and raise the cost. For many existing dwellings heat pumps of the order of 10kW output would be required to meet the space and water heating load. When considering new low energy housing, and also new houses with improved insulation however, a range of output between 5 and 8 kW would probably be required.

Thus it seems likely that the initial input in the domestic heating market could be in larger, private dwellings or by forward looking authorities adopting a large scale housing scheme. The building industry and heating engineers however, are notoriously conservative in outlook and this, coupled with a general ignorance of heat pumps, both amongst consumers and the tradesmen who have to install and maintain them, will contribute to slow entry into the market. To remedy this a large scale energy education programme must be started to inform both potential users, and the trade, of the great potential waiting to be uncovered [4].

Nevertheless, the most overriding factor to affect the wide scale adoption of heat pumps will be their cost. From the consumers point of view, the heat pump will always cost more than a conventional heater and therefore it must be shown to be cost-effective. I will deal with this in more detail in Section 4.2. For the moment, if we consider the cost of a gas boiler, depending on size, to be between £75-200[†], then an acceptable price for a heat pump might be expected to be around £200-£300. At present however, it seems some time before this cost target will be met (especially since the likely cost of electric heat pumps for domestic heating is around £800) and also a considerable amount of development work is still required. Thus I would suggest that it will be quite some time before substantial numbers of gas engine driven heat pumps are widely used in houses. So the indications are that another sector may be the first to

† N.B. These are 1981 prices.

herald a major penetration of heat pumps.

4.1.2 Commercial Sector

Heat pumps are already fairly well established in this market and have been for quite some time, but mainly in Europe and America. Commercial and public buildings have many diverse requirements when it comes to heating [5]. In some cases they may lend themselves to combined heat and power schemes, or need heated water ring mains, heating plus air conditioning, heating plus cooling, heat recovery and so on.

In situations which require one part of a building to be cooled, whilst another area requires heating and/or hot water e.g. refrigerated cabinets, cold stores in shops or in a situation where there is excess heat in one area and not enough in another e.g. high level of occupancy, high level of lighting, solar gain etc. then a heat pump is particularly suited. In such cases the capital cost of the heat pump would compare favourably with that of a heating system plus an air conditioning system, but of course the annual energy costs would be lower. Published examples of such systems include shops, offices, schools and hospitals [6, 7, 8, 9, 10].

The latter, along with nursing homes and hotels, provide a particularly good application for heat pumps, since there are requirements for heating, cooling and heat recovery from waste water (laundry, washing and bathing etc). This gives the possibility of combined heat exchange and upgrading; and

the pattern of continuous occupancy gives rise to a steady load on the heat pump and thus a good efficiency. The heating and cooling of leisure centres, sports halls and swimming pools also offers considerable scope for heat pump techniques [11, 12, 13, 14]. In swimming pool applications heat pumps can be used to recover heat from exhaust air, dehumidification or pool water heating, and in the case where a sports complex has an ice rink as well as a swimming pool, a heat pump may provide pool heating and ice rink freezing from an integrated system (15).

The disadvantages associated with gas engine driven heat pumps in the domestic sector, namely noise and maintenance, pose less of a problem here. Most commercial premises either already have a boiler room or could easily install one, say on the roof. Also regular maintenance can be provided with comparative ease.

The present gas engine driven heat pump system described in Chapters 5,6 and 7 is capable of an average output of 15kW. This output is too large for the majority of individual houses. The capacity was primarily determined by the size of the engine which is the lowest rated engine available with both water cooling and forced lubrication (for good waste heat recovery and reduced maintenance respectively). Thus there are at present, problems associated with scaling down to domestic sizes, due to the lack of suitable engines, however there are no such problems with scaling up. Indeed there are distinct advantages to be gained by scaling up

since larger engines are more reliable and efficient and easier to maintain.

It is therefore apparent that in the Commercial sector heat pumps should be considered for application whenever there is a need for heating, combined heating and cooling, or heat recovery. The cost-effectiveness will obviously depend on the particular application but, in general, the steadier the heat load and the better matched the heating and cooling requirements, the more economic is the system.

4.1.3 Industrial Sector

The application of heat pumps in industry can be classified as follows:

- a) Refrigeration or chilling with heat recovery for space heating
- b) Heat recovery from industrial effluents
- c) Drying of moist materials.

In all industrial applications noise and maintenance should cause no worries, but reliability will be a very important factor. In common with simple heat recovery techniques, it is often found that driving forces, other than the value of the energy savings, contribute towards the investment decision. Such factors as water savings, achieving cooling in addition to heating and improving product quality and process control, can play a decisive role [16]. Let us now look at the three application areas in turn:

a) Refrigeration or chilling with heat recovery for space or water heating

There are a large number of application areas where refrigeration is the primary function of the heat pump/refrigeration cycle. With any refrigeration process, heat from the condenser must be dumped, conventionally via a cooling tower, or directly to the atmosphere. This comparatively low grade heat may be used for preheating or space heating in a location near by. Alternatively by replacing the conventional condenser arrangement with one suitable for heat recovery, which is relatively inexpensive, a very short capital return time can sometimes be demonstrated where savings in fuel cost are significant.

A good example of this can be found at a plastics moulding factory at Telford [5] where a single water chiller is arranged as a heat pump. The chiller provides water at 7°C to the moulding machines and the water heated by these machines is passed to a large underground water storage tank. Heat extracted by the chiller is circulated to fan blown heat pump condensing units, providing heat to the factory in winter. Not only is there a large saving in heating fuel cost, but productivity and product quality are both improved due to the steadier and lower cooling water temperatures, water loss has been virtually eliminated, and heating system safety has been increased following the replacement of fuel-fired units. The non optimised heating mode of such a system will often result in a low COP, but since the energy

consumption is for another purpose, the COP is of less significance.

b) Heat recovery from industrial effluents

There are some processes which require heating and cooling or dehumidification simultaneously, on a single production line, and it is therefore possible to pump heat directly from one part of the process to another. The main areas of potential are in those industries which use large quantities of water as part of the process. Examples are in the food and dairy industries, paper, textile treatment plants, chemical industries and metal finishing. Payback times of 5 years have been quoted, but even this may be too long for industry even allowing for the potentially large energy savings. Some form of government incentives would appear to be necessary to encourage large scale adoption of such systems.

One scheme which is going ahead however, is that of a gas engine driven heat pump in a malt kiln in Grantham. In this case the heat pump is being used to increase the degree of heat recovery from a simple system already in use. During the latter stages of malting it is common practice to re-use the dry exhaust from one kiln to supplement the firing of another kiln in an earlier stage of malting. In 1979 the owners installed a run-around coil system to recover sensible and latent heat from the kiln exhaust to pre-heat inlet air to the kiln. Now, in an effort to recover additional heat from the kiln exhaust, and to move away from direct firing to indirect firing, a heat pump system is being installed to

work in conjunction with the existing system.

The existing exhaust heat exchanger is retained and an additional heat exchanger, the evaporator, will be installed. The output from the heat pump, including the engine exhaust and cooling jacket is used to preheat the air entering the malt kiln. During the malt curing phase, when an increased air temperature is required, a supplementary gas boiler brings the inlet temperature up to 95° C.

The expected payback period is 6.3 years with a target saving of 640 tce/a*. [17]

c) Drying of moist materials

Drying processes lend themselves very well to the use of heat pumps since heat can readily be extracted from a wet exhaust system, thus condensing out moisture and providing a stream of dry air which can be reheated on the output side of the heat pump and recirculated through the drier.

Drying is carried out in nearly every industry and accounts for almost half of the heat used under 120°C. Table 4.4 shows the water removed by evaporation in some of the major industries. [18]

The Watt Committee [19] judged the total of 17.4×10^6 tonnes of water removed to be closer to half the probable total figure. So given 30 million tonnes, and therefore taking an overall drying efficiency of 5% (currently achieved by

* tonnes of coal equivalent per annum

steam), the total primary energy requirement amounts to 211×10^6 GJ per annum, i.e. 2.5% of the UK energy requirement.

Thus the potential savings to be made are clearly worthwhile.

Table 4.4 Drying needs in some of the major industries

Industry	Annual Production $\times 10^6$ tonnes	Average Moisture % Content Drop	Water Removed $\times 10^6$ tonnes
Paper and Board	4.6	200	9.2
Bricks	15.7	15	2.4
Milk-dried	0.21	900	1.85
-condensed	0.17	500	0.85
Gypsum	3.7	20	0.74
Plaster & Plasterboard	2.3	45	1.0
Textiles	1.4	30	0.4
China Clay	3.5	10	0.35
Fertilisers	4.0	3	0.12
Timber-softwoods	0.27	45	0.12
-hardwoods	0.24	20	0.05
Dyestuffs	0.1	50	0.05
Vitrified China Clay Pipes	0.75	15	0.11
Tiles, Pottery and Sanitary ware	1.0	15	0.15
Total			17.4

4.2 Cost-Effectiveness

There are a number of factors which combine to decide if the increased cost of installing a heat pump system will be balanced against the savings expected in running costs. For instance, it will depend on the capital cost of the system, the price of fuels, the fuel that is used for the heat pump, interest rates, complexity of installation etc.

Because of the size of engine problem, mentioned earlier, the most sensible use for heat pumps at present would be for supplying a larger heat or hot water load, conventionally supplied by large boilers or groups of boilers. The applications which immediately come to mind are grouped heating schemes, shops, offices, schools or hospitals.

There are distinct advantages to be gained by aiming for larger machines:- larger engines are more efficient, and problems such as noise and maintenance become less significant since such applications mostly already have boiler rooms, regularly maintained.

The 15kW heat pump system described in Chapters 5, 6 and 7 would cost around £2,500 to build (1980/81 prices.) Mass production would reduce this cost to, say £2,000. Using the results of this work, we can scale up and estimate the cost of components for a larger scale model, capable of say 100 kW output. This works out to around £6000 (i.e. £60 per kW). This is because the costs involved with the production of larger heat pumps will be subject to scale economies: (small cost increase with increasing engine size), and the availability of larger, cheap, mass-produced car engines. Installation costs should be no different from those of a conventional boiler system, given that the heat pump would probably be manufactured as a packaged unit. Maintenance costs however are likely to be higher, due to the refrigeration techniques which must be made available, and engine servicing, say £200 per annum as compared to £100 per

annum for a gas boiler of a similar size (100 kW).

Running Costs: (1980 prices)

- a) 100 kw gas boiler (75 % efficient) = 1.209 p/kWh useful
- b) 100 kw gas engine heat pump = 0.756 p/kWh useful
(C.O.P. of 1.2)

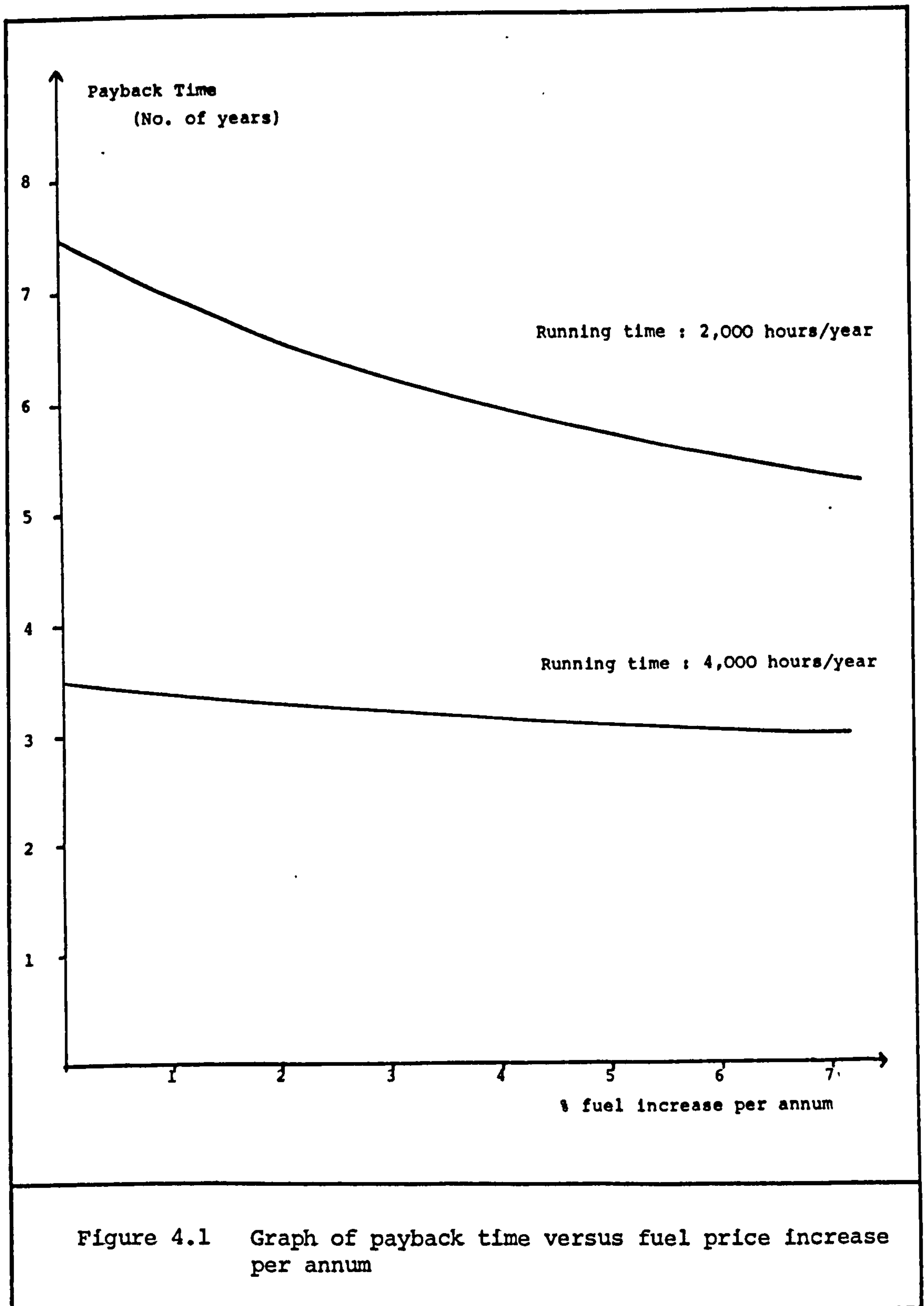
If we now assume a typical running time of 2000 hours per year:

- a) annual cost of running gas boiler = £2418 per annum
- b) annual cost of running heat pump = £1511 per annum

Therefore the savings per year are £2418-£1511 = £907, minus the difference in maintenance costs (£100) = £807

Hence assuming that fuel prices remain constant, this gives a simple payback time for the heat pump (based on the work detailed later) of 7.44 years. The results of assuming real fuel price increases per annum, plus an increased running time of 4000 hours/year in order to simulate a higher load factor situation, can be seen in Figure 4.1. overleaf.

A more detailed analysis, enabling a comparison to be made of the two systems incurring different expenditure patterns over time, can be carried out using the method of discounted cash flow analysis. This evaluates the net present cost (NPC) of a flow of expenditures by discounting future expenditures back to the present, using an appropriate rate of discount.



If we assume that the annual running costs i.e. maintenance and repair of a heating system are constant in real terms, and that any rise in fuel prices takes place smoothly, then the NPC of any heating system is given by:

$$NPC = C_H + R_f \sum_{t=1}^N \frac{(1+f)^t}{(1+i)^t} + R_o \sum_{t=1}^N \frac{1}{(1+i)^t}$$

where

C_H = capital cost of heating system in year 0

R_f = annual fuel cost (present values) i.e. year 0

N = system lifetime (No. of years)

f = annual rate of growth of fuel prices

i = annual rate of discount

R_o = annual running cost in year 0 (maintenance etc.)

There are a number of assumptions implicit in this expression:

1) It is assumed that the technical performance of the system is constant over the N years.

Given that the greatest effects will be associated with ageing i.e. achieving lower performance in later years, the effects of discounting reduces this problem since the effects in later years become less significant.

2) It is assumed that real fuel prices rise or fall at a constant annual rate over the N years.

This variable has a major effect on any calculation, so its value must be chosen carefully. However to estimate future fuel prices, and the rate of change, one needs slightly more

than a crystal ball.

There is at present a widely respected viewpoint that real fuel prices will double by the end of the century, and this can be achieved in the equation by putting f equal to 0.04, i.e. a 4% per annum rate growth of fuel prices. This is probably the maximum rate of increase that could be sustained.

The other major item for consideration in the equation is the rate of discount to be used which is one of the tools of monetary policy. The discount rate applied to the public sector in the UK is laid down by the treasury, and at present (ie 1981) stands at 5%. A higher discount rate of 10% is used to represent the industrial sector. The discount rate thus represents the rate of interest charged to borrowers by commercial banks and other financial institutions. The NPC formula can be simplified as follows:

$$\text{NPC} = \underbrace{C_H}_{\substack{\text{Capital} \\ \text{cost}}} + \underbrace{x R_f}_{\substack{\text{Annual} \\ \text{fuel} \\ \text{cost}}} + \underbrace{y R_0}_{\substack{\text{Annual} \\ \text{running} \\ \text{cost}}}$$

$$\text{where } x = \sum_{t=1}^N \frac{(1+f)^t}{(1+i)^t} \text{ and } y = \sum_{t=1}^N \frac{1}{(1+i)^t}$$

Overleaf is a table showing four different NPC cases used for comparison.

Table 4.5 Summary of Parameters used in the NPC Calculations

Parameter	NPC (1)	NPC (2)	NPC (3)	NPC (4)
f	0	0.04	0	0.04
i	0.1	0.1	0.05	0.05
N	20	20	20	20
x	8.5	11.7	12.5	18.1
y	8.5	8.5	12.5	12.5

NPC(1) uses a constant fuel price over 20 years, with a 10 % discount rate.

NPC(2) again uses a 10% discount rate, but with a 4% per annum rate of growth of fuel prices, over 20 years.

NPC(3) uses a constant fuel price over 20 years, but with a lower discount rate of 5%.

NPC(4) again uses the lower discount rate of 5%, but with fuel prices increasing at 4% per annum, over 20 years.

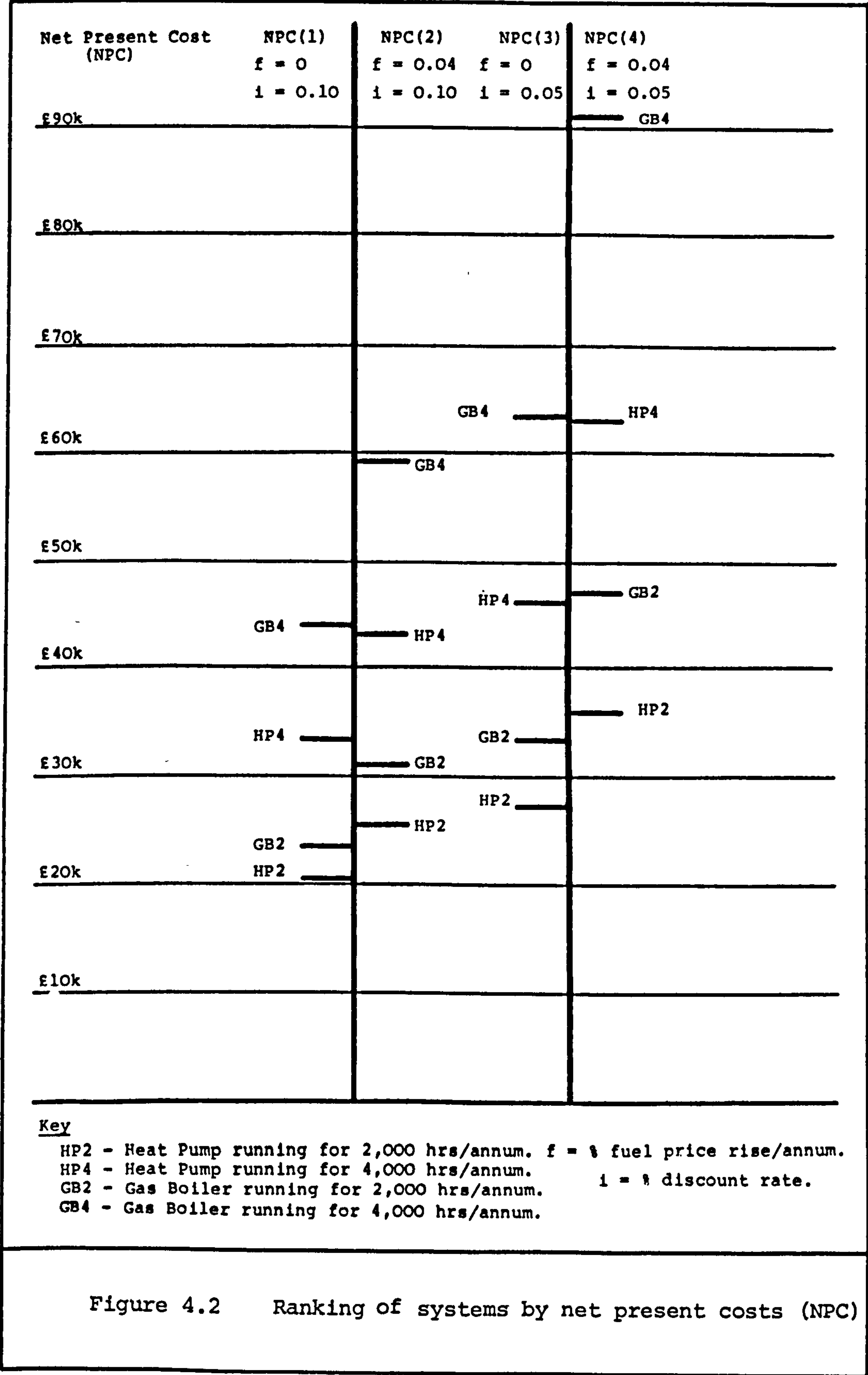
The results of these calculations - the Net Present Costs of the four cases for each of the systems, are shown in Table 4.6 for a typical running time of 2000 hours per year and also for the increased running time of 4000 hours per year (increased load factor).

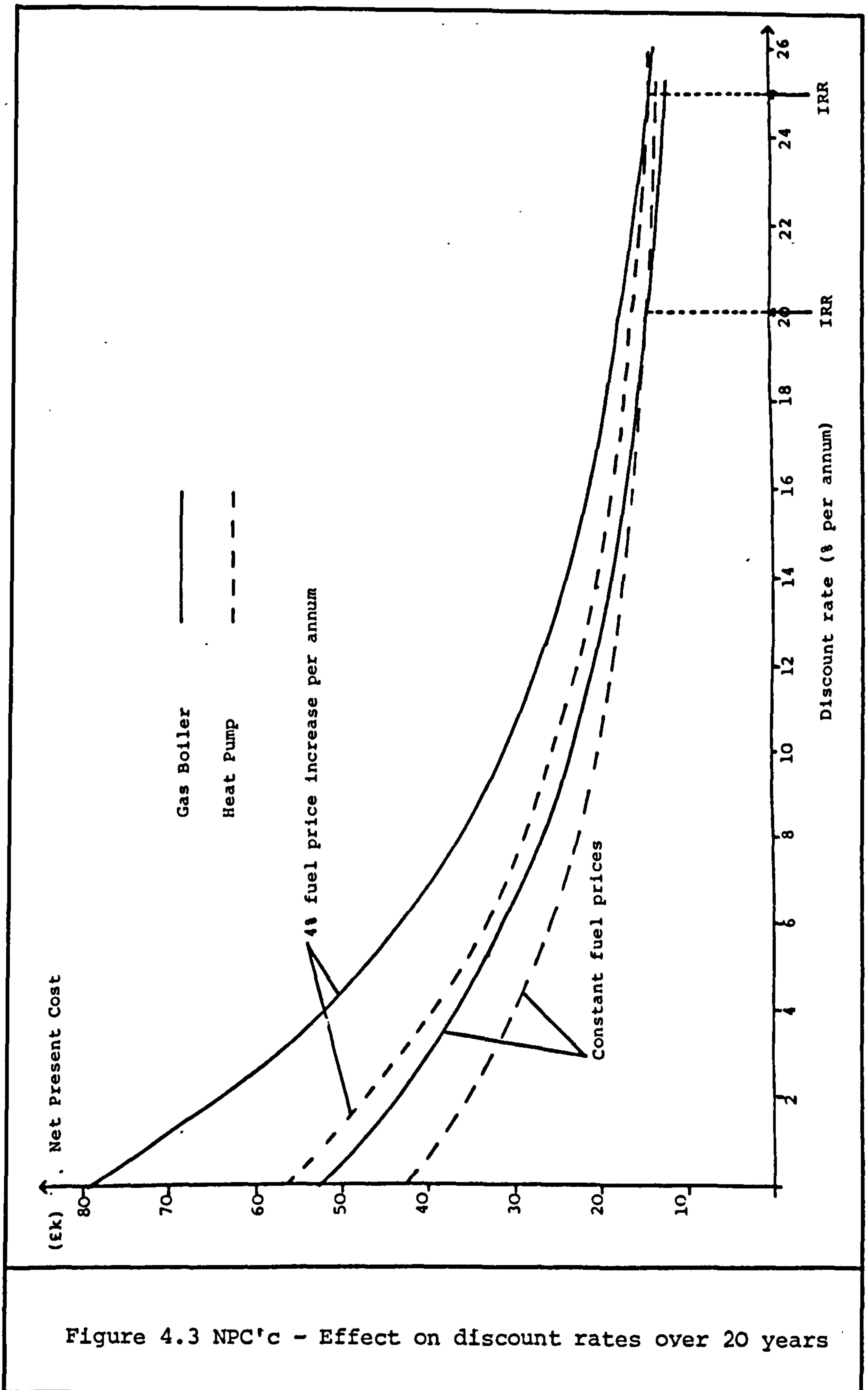
Table 4.6 Results of the NPC Calculations for a Gas Boiler and a Heat Pump. (£k - 1980 prices)

	NPC (1)	NPC (2)	NPC (3)	NPC (4)
2000 hrs running p.a. - gas boiler	23.47	31.15	33.43	47.10
- heat pump	20.60	25.40	27.37	35.92
4000 hrs running p.a. - gas boiler	44.06	59.41	63.56	90.91
- heat pump	33.47	43.06	46.20	63.29

These results can then be used to rank the systems as shown in Figure 4.2. overleaf.

The NPC calculation is very sensitive to variations in discount rates and fuel price rises. This is illustrated in Figure 4.3 on the following page which shows the NPC at varying discount rates for two different cases, one assuming constant fuel prices and the other a 4% per annum growth. The internal rate of return (IRR) is shown as the area of the graph where the line for the boiler and heat pump cross. Thus at constant fuel prices, as the discount rate decreases below 20%, the economic case for a heat pump system, becomes greater. Using the 4% p.a. rise in fuel prices, the IRR increases to 25%, making the system even more cost effective.





CHAPTER 5 OUERG/LUCAS HEAT PUMP - INITIAL DESIGN

CONSIDERATIONS

5.1 Background and Aims of the Project

In 1977 a joint venture was embarked upon by the Open University Energy Research Group (OUERG), Lucas Aerospace (Burnley) and Milton Keynes Development Corporation (MKDC) to design, construct and install a domestic heating system, based on a gas engine driven heat pump, in a house in Milton Keynes. It was envisaged that the system would be fully monitored for a year before the house was rented in the normal way, and would then be monitored for a further year to determine the effect of occupancy on the system.

The initiative for the project came from the Open University's Energy Research Group [1] (ERG). ERG was established in 1972 and its many activities cover a wide spectrum of energy and energy related issues. One of the major fields of research was, and still is, in energy conservation. There are two ways of using energy more efficiently - one is to require less (eg. by insulation) - the other is to do the same with less (ie. improve appliance efficiency). A great deal of work had already been done or is underway on the first method, including the monitoring of an estate of low energy, passive solar houses and the production of a short course for the public entitled 'Energy in the Home'[2]. The work described in this thesis concentrates on the second method of using energy more

efficiently, by improving on appliance efficiency.

Dr R P Bush [3] of the Energy Technology Support Unit, Harwell, has drawn up a table similar to Table 4.1 in Chapter 4, which summarizes the efficiencies of various alternative space heating systems and this is shown below as Table 5.1.

Table 5.1 Efficiencies of Primary Fuel Utilisation in Various Heating Systems.

	Primary Fuel Input	Energy deliv. to Consumer	Input to Heat Pump	Output @ $COP_H = 3.0$	Waste Heat Recovery	Total Output
Electric Resistance Heating	1.00	0.29	-	-	-	0.29
Direct gas Heating	1.00	0.94	-	-	-	0.75
Electric Heat Pump	1.00	0.39	0.29	0.87	-	0.87
Gas Engine driven H.P. with heat recovery	1.00	0.94	0.28	0.84	0.33	1.17

Thus it is clear from the table that fossil fuel fired heat pumps offer the prospect of much more efficient use of primary fuel than any of the alternatives mentioned.

The Milton Keynes Development Corporation is one of the few authorities still with a large building programme and considers that it has a responsibility to encourage and try out new ideas, which, if repeated on a national scale, could either reduce energy demand or increase energy supply. To this end it set up an Energy Consultative Unit in 1976, working in close liaison with the OUERG [4, 5]. Several

joint projects were undertaken, the first being a flexibility study showing the effect of increasing fuel prices on the new city of Milton Keynes. Two major projects (still in operation) are to monitor and analyse low energy houses. In one project the fuel consumption and energy flows in 177 houses of various design is being monitored and statistically analysed to determine the effects of different levels of insulation and of passive solar energy techniques on domestic fuel consumption. The other project consists of eight highly insulated passive solar houses, monitored to an extremely high level. The aim is to examine in detail the patterns of energy use and heat flows through houses [6]. Therefore when a gas engine driven heat pump was proposed at the Open University, MKDC felt it fitted in with their overall plans and offered a property as a test bed for the system.

The project partnership was complete when Lucas Aerospace expressed an interest in the construction of the heat pump. Faced with declining orders and a large cut in their workforce, the trade union members of the company were looking to diversify their range of products. Several ideas were put forward such as kidney machines and heat pumps, but the management showed little enthusiasm. Lucas were not the ideal company to undertake such a project - they were geared to high precision aircraft engineering. However, they did have a long history of involvement with heat pumps and indeed had manufactured them in the 1950's [7]. Also their experience with internal combustion engines and wealth of engineering knowledge would complement the energy system

design work at the O.U. and the heating and ventilating experience of engineers at MKDC. With the promise of government backing the Lucas management were finally persuaded to join the project and provided a development engineer, a machinist and a fitter to build the unit.

The main aim of the project was to design, construct and monitor the performance of a gas engine driven heat pump. To collect data and gain valuable experience of the difficulties associated with the operation of gas fired heat pumps with a view to the design of a larger commercial unit which is believed to be a more economically viable device at the present time (due mainly to the lack of small efficient engines). In particular it was aimed to verify the demonstration design assumptions by monitoring the heat pump under a wide range of operating conditions and to investigate the effect of different control strategies. Another valuable piece of information sought, related to the maintenance and reliability of engine driven systems and of noise levels, which effect the widespread acceptability of such heat pumps.

5.2 System Design

From a computer analysis, taking into account the house design, constructional materials and the required levels of temperature in different parts of the house, it was established that the house would require 6.2 kW of heat at an ambient temperature of 0°C, assuming 1 air change per hour. These figures were taken as the basis for all the design calculations. (Further details and a listing of program are

given in Appendix 1.)

As previously discussed in Chapter 3, there are three major choices to be made when designing any heat pump heating system. A choice of

- a) prime mover
- b) heat source and
- c) distribution medium.

Each has to be carefully considered in turn.

5.2.1 Choice of Prime Mover

As I have already stated, most work on heat pumps to date had been confined to electric drive units. Such units are well documented and consist mainly of modified refrigeration units optimized for reliability. They are the subject of many research programmes trying to improve on their energy efficiency since, in terms of primary energy, they suffer a low energy conversion efficiency due to the losses incurred in the generation and transmission of electricity. A fossil fuel driven system, on the other hand, avoids these losses at the power station, and also further increases its heat output by utilizing any waste heat gained from the engine.

In using such a system however, there are major uncertainties which need to be answered, namely questions of noise, maintenance and reliability, all of which the project set out to answer.

An internal combustion engine was chosen, rather than one of

the more exotic external combustion engines, as it was seen to have three advantages:

- i) the experimental unit could be built quickly and cheaply, and could provide much general information on engine driven heat pumps
- ii) the i.c engine promised to be as efficient as any of the available alternatives
- iii) the commonly quoted disadvantage of i.c. engines are noise and excessive maintenance requirements. We believed that it would be possible to overcome these problems and so take advantage of the low capital cost of an engine type already in mass production.

Initially it was planned to use a liquid fuel for the engine, however the associated storage costs, larger fire hazard and more corrosive and toxic emissions persuaded us that it would be better to use natural gas. After an exhaustive search of manufacturers and suppliers the only engine suitable was a single cylinder 360cc Brit imp marine petrol engine, capable of delivering 3 kW of shaft power. Although still too big for our needs this was the lowest rated engine available with both water cooling (for good waste heat recovery) and forced lubrication (for reduced maintenance).

In an average heating season the heat pump would be expected to run for some 800 hours, which is equivalent to a car travelling 24,000 miles at 30 m.p.h. Using Meteorological Office data over a 27 year period a computer program was written to determine how long the heat pump would have to operate under various conditions (see Appendix II). The

results are shown in Table 5.2 below which shows average running times of the heat pump under different ambient temperatures in bands of 2°C.

Table 5.2 Heat Pump Operating Hours under Various Ambient Conditions

Ambient Temperatures °C	No. of Hours Operating
Below -4.0	1.9
-4.1 to -2.1	6.6
-2.1 to 0	25.3
0.1 to 2.0	52.9
2.1 to 4.0	87.2
4.1 to 6.0	105.0
6.1 to 8.0	128.0
8.1 to 10.0	155.0
10.1 to 12.0	108.0
12.1 to 14.0	81.8
14.1 to 16.0	40.9
Above 16.0	32.2

In order for the heat pump to perform well with the minimum of maintenance it was proposed to investigate methods of increasing engine reliability however the project was curtailed before we could assess this fully. Some of the ways we hoped to achieve this increased reliability were by:

- a) stellite valves and seats
- b) modifying ignition and timing.

Also, reliable starting could not be ignored and this too had to be investigated.

5.2.2 Choice of Heat Source

The various factors which can affect the choice of heat source were discussed earlier in Chapter 3 but we were also limited by the fact that we wanted our choice of source to be applicable to a wide range of housing in Milton Keynes, a factor which ruled out a number of contenders. In Milton Keynes there are no industrial processes or power stations where surplus waste heat or water can be utilised. Ground coils were discounted as they would occupy too much space on large housing estates and are inaccessible for maintenance and repair. Most dwellings are not conveniently situated to a large body of water or a flowing stream or river.

The most abundant and easily accessible heat source in Milton Keynes appeared to be the air although this choice has several disadvantages associated with it. As is the case throughout the U.K. a damp atmosphere prevails which encourages the formation of ice on the evaporator coils at low temperatures. (If air is cooled below its dewpoint the moisture in it condenses and may freeze on the heat exchanger thus forming an insulating layer and preventing the efficient conduction of heat from the air to the heat pump). This problem is usually overcome by designing the air flow over the heat exchanger to be as large as possible (minimising the temperature drop in the air) and by using a suitable defrost mechanism. The other major disadvantage with air is that its variability of temperature is considerable, leading to a variable C.O.P., and this was shown in Figure 3.14 in Chapter 3.[8].

Despite the disadvantages however, air is the obvious choice for any type of heat pump which is hoping to serve a mass market. Appendix III shows graphs of mean monthly temperatures for each hour of the day over a 25 year timespan from 1949 to 1975. The data is for Heathrow and from a survey of Met. Office statistics it would appear that the temperature in Milton Keynes is generally 1 degree lower than it is in London.

5.2.3 Choice of Distribution Medium

In order to fully realise the potential benefits of a gas engine driven heat pump it is necessary to integrate it into a conventional heating system. A central heating system distributes heat around the dwelling from a central heat source using one of two traditional methods - one using hot water as the heat distribution medium, and metal radiators, and the other using air as the distribution medium, via a network of ducts. There are advantages and disadvantages to using both and the choice was by no means clear cut.

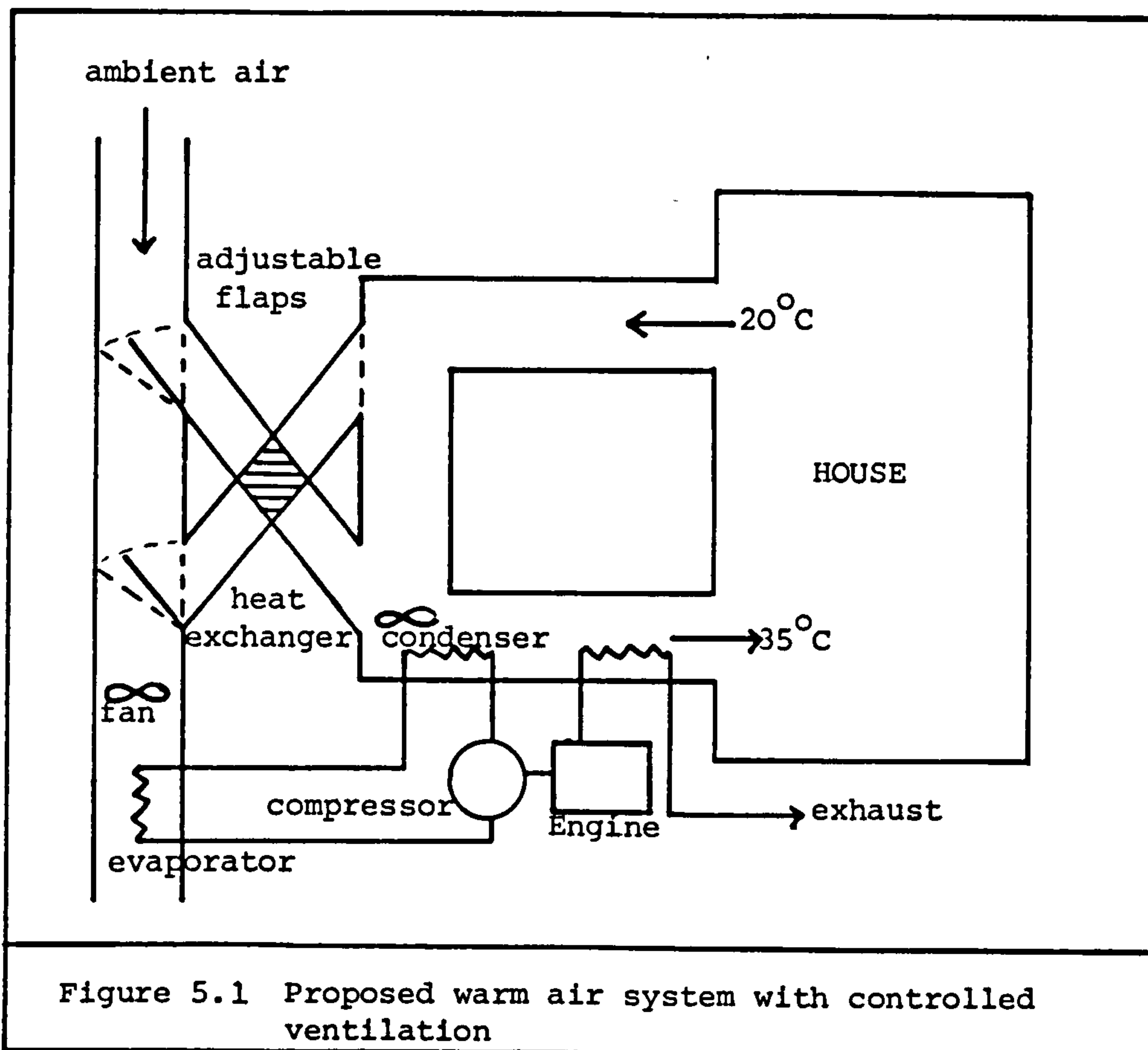
The higher the temperature of the heat distribution medium, the worse the average performance of the system. Water distribution has a practicable minimum temperature of about 60°C, and air distribution about 40°C. Heating by air requires large ducting throughout the house whereas water distribution only requires simple plumbing. At the delivery point, warm air vents take up no space in a room, but large panel radiators restrict furniture layout etc. The water

distribution system can also supply domestic hot water (D.H.W.) but the air system cannot. The air system has a faster response time than the water system but the ductwork is liable to transmit engine noise and vibration. There are many more arguments for and against each system, which makes it very difficult to pick out the "best" one.

The problem was simplified somewhat when constraints on the engine were taken into account. To maximise the engine efficiency it should be run at it's designed output on an intermittent basis, rather than continuously at some fraction of full output. Also a large number of automatic starts per day should be avoided. Thus it seemed sensible to use a thermal buffer store for the delivered heat. Two possible systems were designed to make use of such a buffer store, one using a water distribution system, and the other air, but using the same engine and compressor. Rather than make an arbitrary choice it was decided to proceed with a trial of both systems, staggered by about six months.

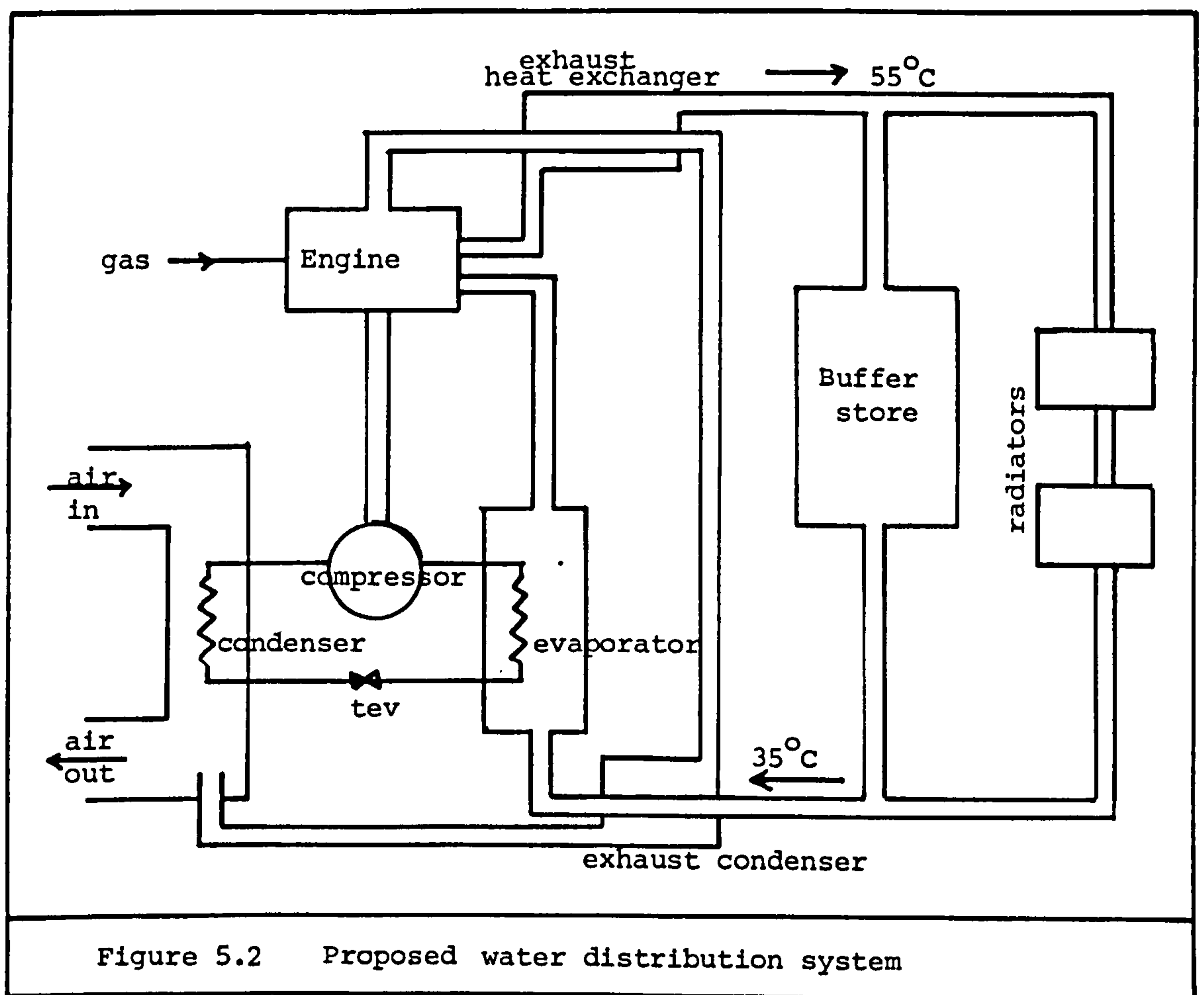
After further consideration, it was decided to omit the buffer store from the air system and use instantaneous operation. Because the heat is low grade (35°C), the size and cost of a water heat store capable of containing a day's heat would be too great and also any storage medium would add to the temperature drop across the heat pump, thereby decreasing the C.O.P. Using the instantaneous approach ventilation recovery can be built into the system at little extra cost enabling the house to be sealed as tightly as

possible, thus minimising ventilation losses. Also, the cycling problem, experienced by electric heat pumps can be overcome since it is possible to modulate the capacity by changing the speed of the engine.



As can be seen in Figure 5.1 the system is basically a variable recirculation rate warm air distribution, with heat recovery by means of a static heat exchanger and heat pump. It is interesting to note that the adjustable flaps can be set such that the minimum ventilation rate is achieved. The control was envisaged to be on/off signalled by a room thermostat. The internal fan is driven from the engine (speed of engine controlled by outside thermostat). As the ambient temperature falls, the speed of the fan and engine increases to give greater heat output. This was to be designed to give long cycle times.

The water system shown in Figure 5.2 was designed to replace a conventional gas boiler system hence there were a number of constraints imposed upon the design, the major one being the supply and return temperatures into the house. We limited ourselves to using conventional equipment and not taking up more wall space with radiators than conventionally acceptable. This meant a supply temperature of 55°C.



The output water can either fill the buffer store or heat the house via the radiators. The buffer store has a capacity of 45 kwh in two 1m³ tanks, although their actual capacity was slightly less than this due to added insulation inside the tanks. When the buffer store is full the heat pump ceases operation and the

radiators take their supply from the store until it is emptied. By running once a day, in the afternoon when the external air temperature is higher, the C.O.P. can be increased. This also minimises any noise problem.

Both systems offered substantial savings. The annual house heat demand was calculated to be 41.8 G.J. This could be satisfied by a convential gas boiler system using 80.45 G J. of gas but the calculations indicated that by using an air to water gas fired heat pump, gas use could be reduced to 29-31 GJ and to 29-33 GJ for the air system. The two different approaches were seen to generate entirely different engine running characteristics, enabling us to appraise the reliability of engines running in different modes. Unfortunately, due to a delay in receiving government funding, we were unable to go ahead with both systems and could only proceed with the water distribution system. The advantage of this system being that a direct comparison could be made with the existing gas boiler already installed in the house.

5.3 Computer Model of System

In order to evaluate the design of the heating system it is important to know how it will behave in the house prior to installation and therefore some sort of model is needed to simulate the thermal interaction which takes place between the house and the heating system.

The thermal response of the house and the performance of the heat pump are both functions of ambient conditions and therefore, given varying meteorological data, a computer model can simulate the relationships. This is particularly important when trying to determine the optimum control strategy for the system. A simplified flow chart for the model is shown overleaf in Figure 5.3.

In order to set up the simulation it is necessary to have house constructional data to determine its rate of heat loss, heat pump design calculations, plus an initial set of weather data. The weather data is taken from a meteorological office for computer tape for Heathrow giving data over a 27 year time span. The heat pump design conditions are based on manufacturers data.

There is a heating demand programmed between 6 am and 9 am in the morning and between 4 pm and midnight at night. Also, there is a hot water demand at 8 am, 1 pm, 7 pm and 9 pm. The heat pump itself always operates between 1 pm and 5 pm, provided the store requires it (i.e. bottom temperature less than 55°C). It is also on when the heating system is on and there is a heating demand not met by the buffer store. A simplified flowchart of the heat pump simulation is also shown, in Figure 5.4. It was hoped to test and refine the program had we installed the heat pump in an actual house, nevertheless we were able to gain valuable information on the size of the buffer store that we would have needed and how the heat pump would respond to different ambient conditions

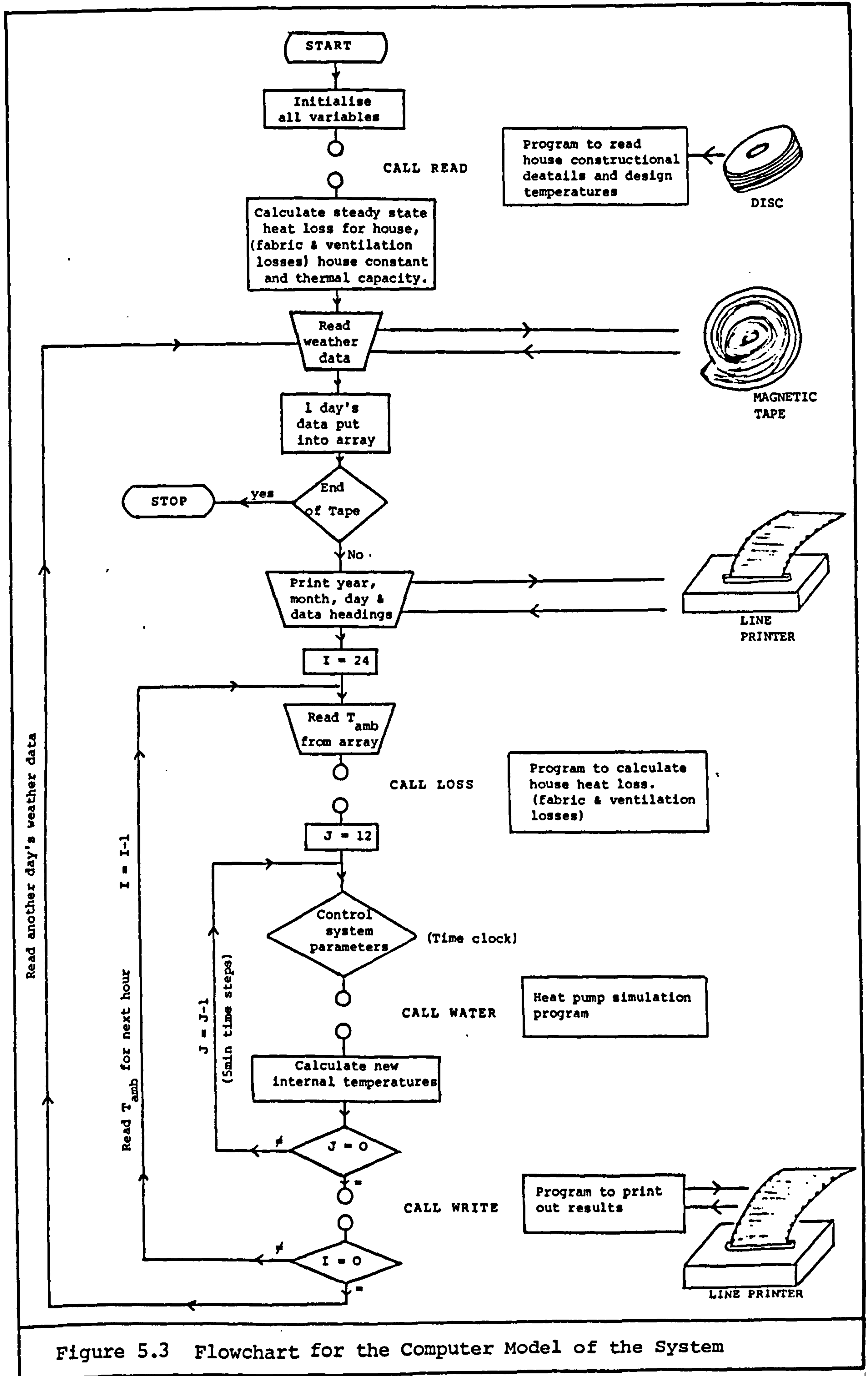


Figure 5.3 Flowchart for the Computer Model of the System

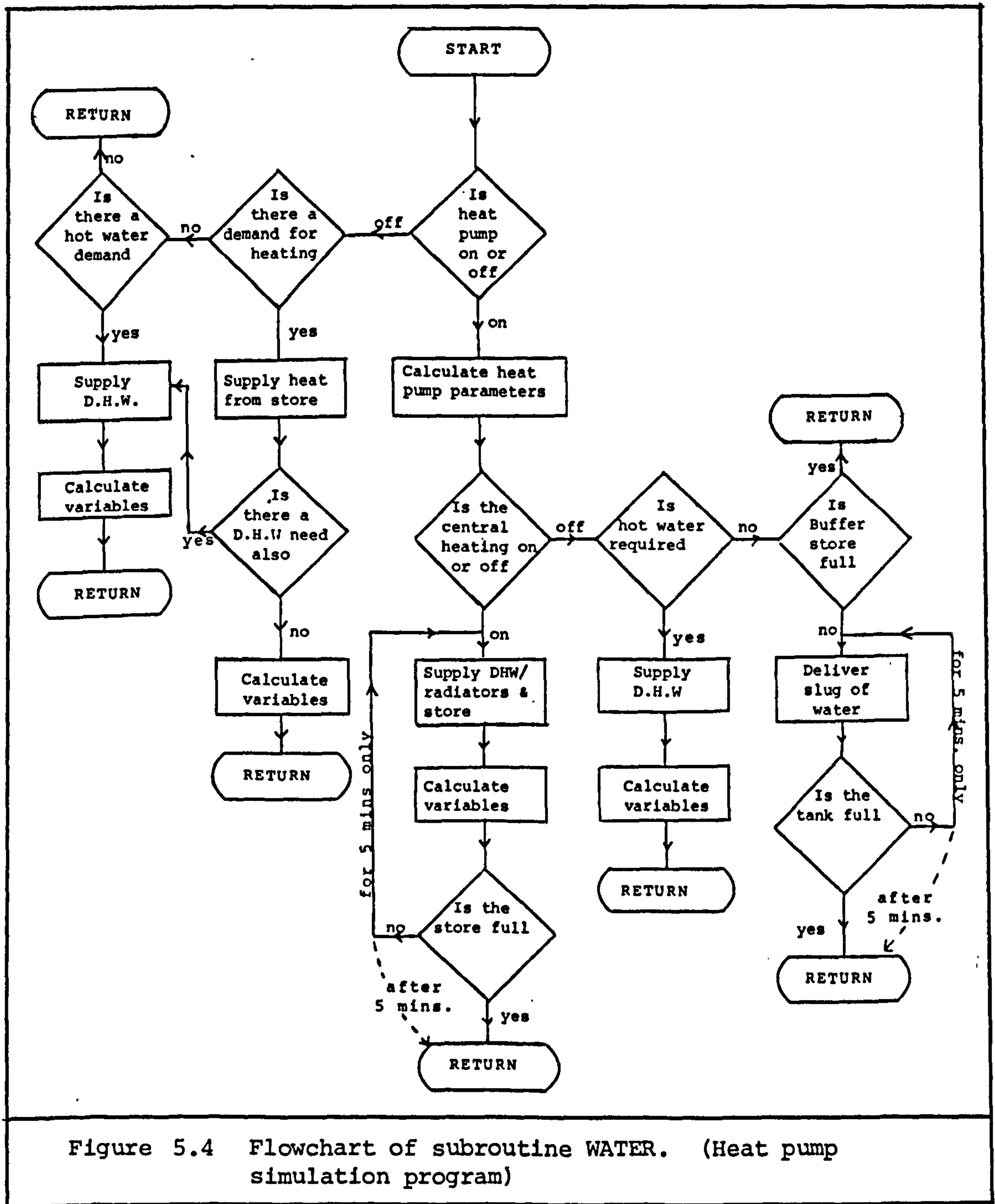


Figure 5.4 Flowchart of subroutine WATER. (Heat pump simulation program)

and the effectiveness of the control system. The output was capable of giving the results for a number of variables in the system for each hour of the day in a year. A complete listing of the program and an example of one full day's running in January is given in Appendix IV on page 210, however outlined below is the state of the system at 6pm at that day with an explanation of the variables that are calculated.

TIME	18.00hrs
TAMB	8°C ambient temperature
KSYS	1 The state of the time clock. 1 = on, 0 = off
BUFEN	46.08kWh useful energy remaining in the buffer store
MDOT3	0 water flow rate to DHW
HOUT	8.31kW power delivered to house in last time step
TI	18.05°C internal temperature
ELUSE	0.81kWh electricity use
GASUSE	62.76kWh gas use
TTANK(1)	57.17°C temperature of water in top of buffer store
TTANK (100)	35.0°C temperature of water in bottom of buffer store
MDOT	0.09 kg sec ⁻¹ total water flow rate
MDOT1	0.09 kg sec ⁻¹ water flow rate to radiators
MDOT2	0 water flow rate to buffer store
DHWCUM	0 cumulative heat delivered to DHW
HCUM	37.37 kWh cumulative heat delivered to house
TC	40.70°C water entry temperature to condenser
TH	51.99°C water exit temperature from exhaust heat exchanger
TENON	6.33hrs cumulative time the engine has run

CHAPTER 6 OUERG/LUCAS HEAT PUMP - CONSTRUCTION AND PROGRAMME OF WORK

6.1 Mode of Operation

6.1.1 Mode of Operation of System

Figure 6.1 overleaf shows a schematic layout of the main components of the system.

Natural gas from the normal domestic supply is piped to the engine driving the heat pump and outside air is drawn in by a fan powered by a belt drive from the engine. The air is passed through the evaporator, cooled and finally rejected. The heat transferred to the freon is released in the condenser, where it heats the inlet water by approximately 10°C. The water then passes through the engine cooling jacket and the exhaust gas heat exchanger before leaving the system at approximately 55°C. Initially the water passes through the exhaust condenser before entering the main condenser.

6.1.2 Mode of Operation of the Heat Pump

Figure 6.2 shows the layout of the refrigeration components of the heat pump. The ambient air, acting as the heat source, is drawn in and over the evaporator by means of a centrifugal fan which is belt driven from the engine.

The evaporator consists of a series of finned tubes, and is fed with liquid refrigerant (Freon 12) from the expansion

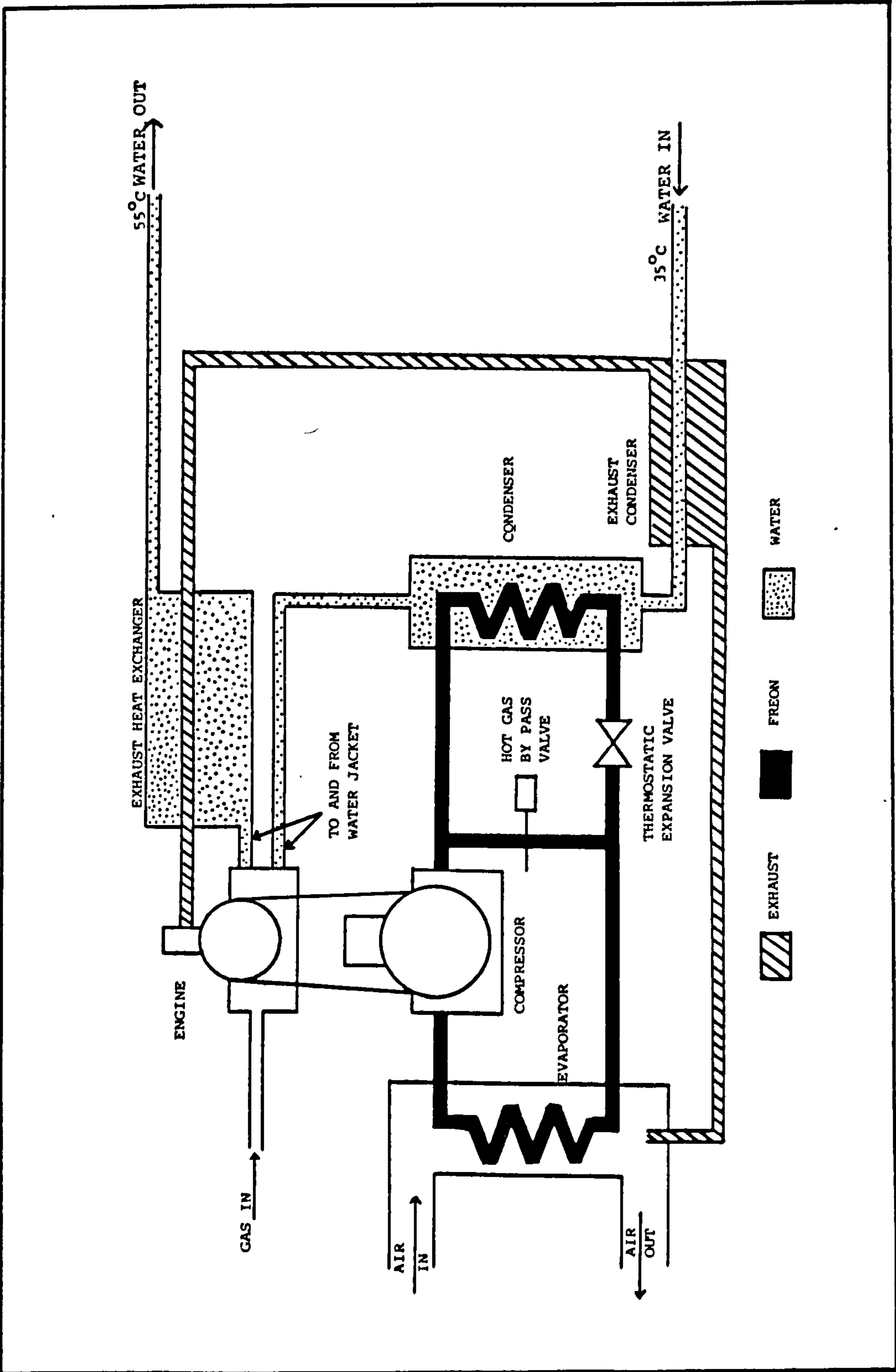
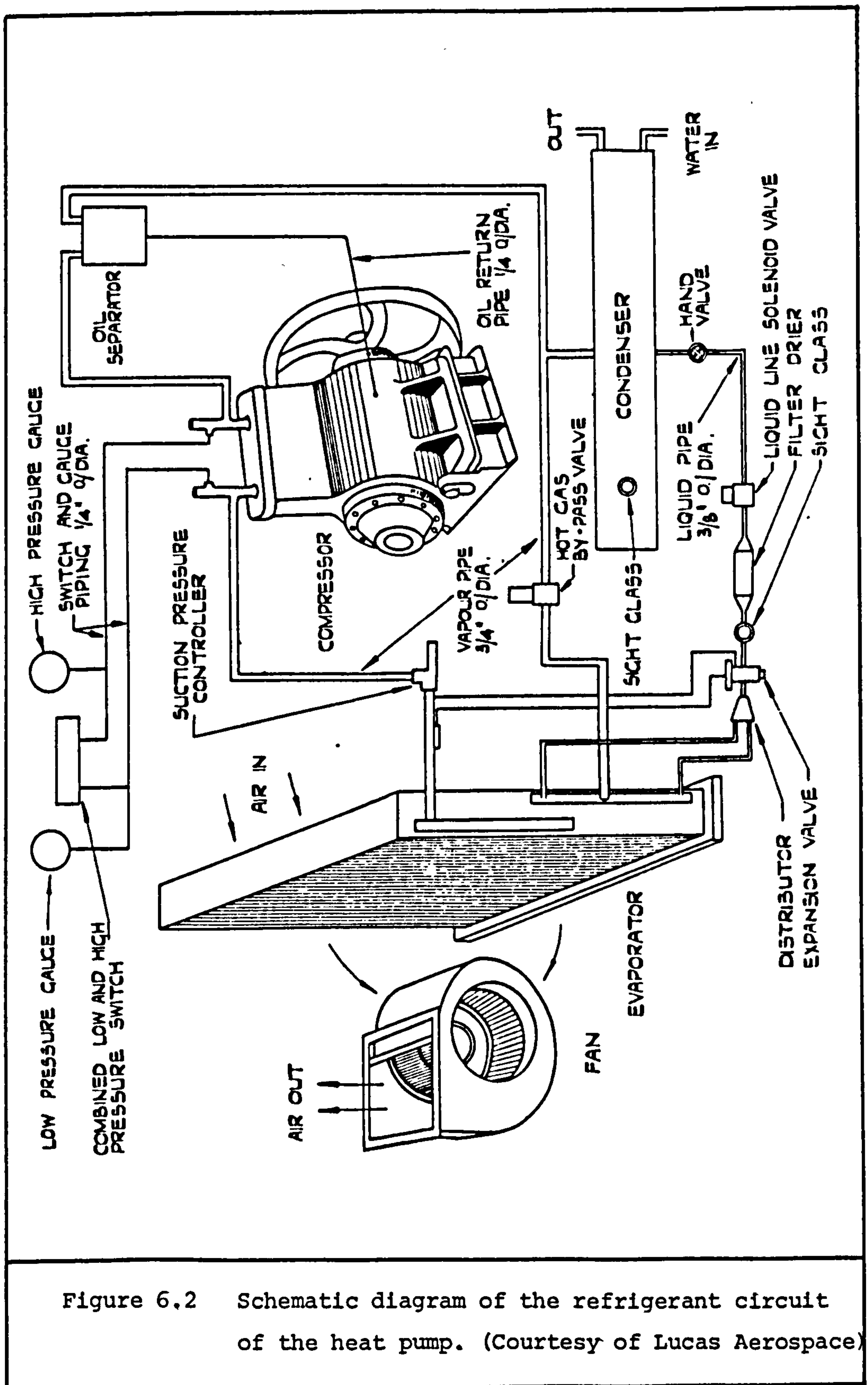


Figure 6.1 Schematic diagram of OUERG/Lucas gas engine driven heat pump.



valve at low pressure and temperature. The refrigerant is then able to take in heat from the ambient air and evaporates. This transfer of heat does not raise the temperature of the freon, but is stored as the latent heat of vaporisation.

The compressor then increases the pressure and temperature of the vapour pumped from the evaporator. Thus the vapour now contains the latent heat of vaporisation plus the heat equivalent of the compressor shaft power (less losses).

The hot vapour is then piped to a condenser where it comes into contact with water cooled tubes, and then condenses to a liquid which collects in the lower part of the condenser until required by the evaporator. As it condenses, the vapour transfers some of its sensible heat and all of its latent heat to the water.

The thermostatic expansion valve regulates the flow of refrigerant to the evaporator to ensure maximum operating efficiency of the evaporator.

The ancillary refrigerant equipment shown in the figure consists of the following:-

- (a) A high pressure switch, which is a safety device connected to the compressor discharge opening at a preset pressure.

(b) An oil separator, which separates out and returns any oil present in the compressor discharge to the compressor crank case.

(c) A hot gas by-pass valve, which allows hot vapour to be fed directly from the compressor discharge to the evaporator. This is the means of removing any build up of frost on the fins and tubes of the evaporator.

(d) A liquid line solenoid valve, which stops the flow of refrigerant to the evaporator during shut down of the system. This is necessary to avoid damaging or overloading the compressor by having a liquid filled evaporator when starting, since liquid refrigerant may enter the compressor causing damage to valves, etc.

(e) A sight glass, upstream of the expansion valve, which shows the condition of the refrigerant, i.e. to check that vapour bubbles are not present, indicating a malfunction of the system.

(g) A sight glass, in the condenser/liquid receiver, which is used to indicate if there is sufficient refrigerant liquid to cover the lower water cooled tubes and provide for sub-cooling.

(h) A distributor, which is part of the evaporator,

and ensures that equal quantities of refrigerant are delivered to each part of the evaporator.

(i) A suction pressure regulating valve, which limits the suction pressure to a maximum preset value to avoid the compressor overloading the engine.

(j) A low pressure switch, which is connected to the compressor suction line, and opens when the pressure falls below a preset value.

6.2 Technical Description

6.2.1 Technical Description of Engine Assembly

Figure 6.3 and Plate VII show the engine assembly. The engine itself is a Brit 'Imp' water cooled, four stroke, single cylinder marine type engine, capable of a maximum speed of 2,000 rpm, and converted to run on natural gas by fitting an Impco gas carburettor (CA50). This was the lowest rated engine available with both water cooling and forced lubrication.

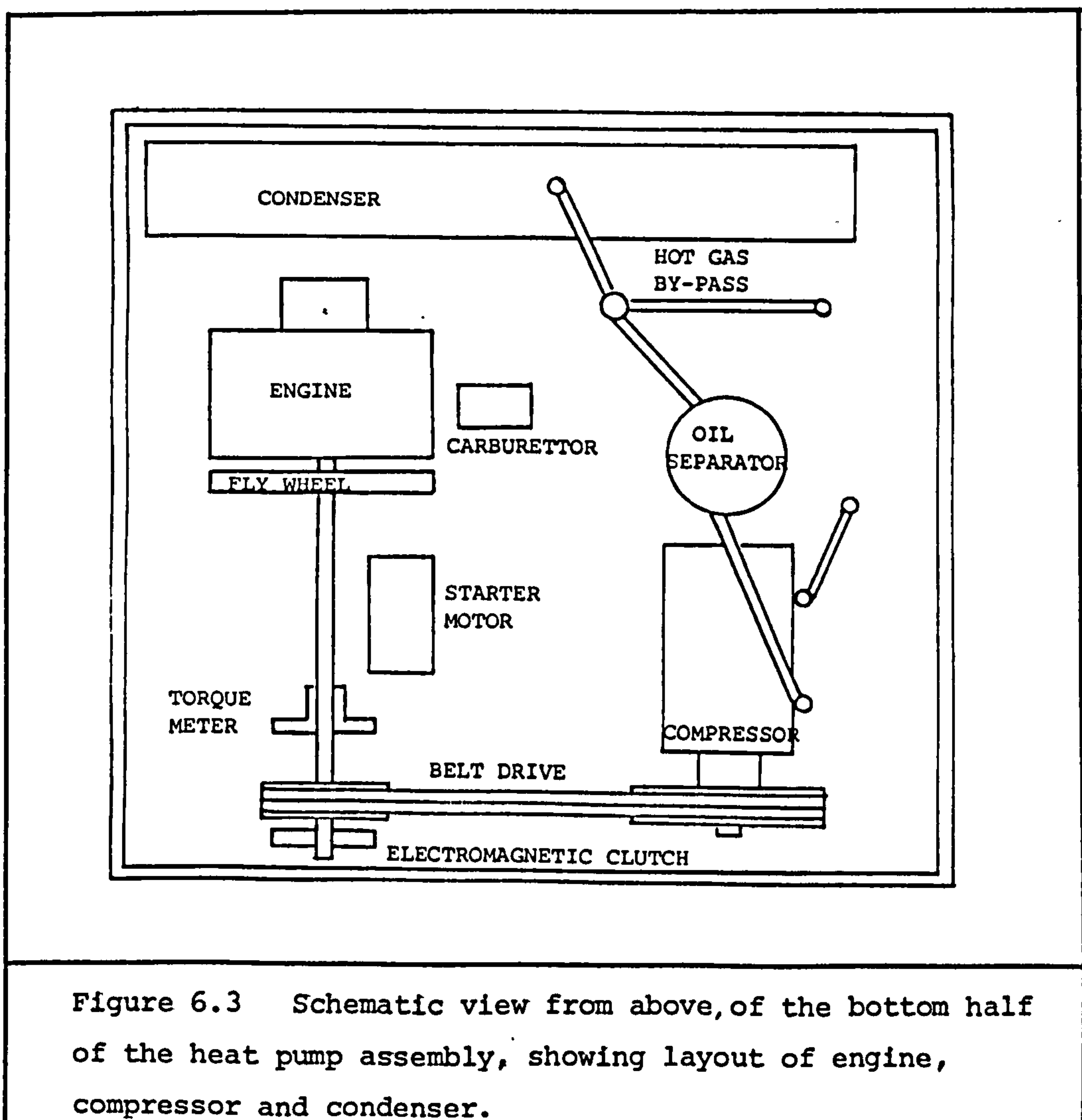
The engine is mounted alongside the compressor, and since both rotate clockwise, the drive is taken from the front or flywheel part of the engine. To start the engine, a standard engine gear ring (10.832 inch bore) is shrunk onto the flywheel and a solenoid starter mounted on the sub frame. A short adaptor shaft bolted to the flywheel carries an electromagnetic clutch to enable the engine to be started off

load. The shaft is also strain gauged, and has a transmitter assembly to enable torque measurement. The shaft is supported by a Fenner type 040C0516 pillar bearing and drives a belt by means of a Fenner taper lock pulley. The sub frame is constructed of 25 mm square section tubing and 50 x 12 mm mild steel strip, with dowels inserted to maintain alignment.

6.2.2 Technical Description of Engine and Condenser Package

The remaining components in the bottom half of the unit are as follows:-

(see Figure 6.3 and Plates VIII and IX).



The compressor is a Hall Thermotank compressor, type 3C. At its maximum speed of 1000 rpm it absorbs 2.4 kW of power when the evaporating and condensing temperatures are -5°C respectively. It is fitted with the normal back seating service valves and a twin belt drive pulley having a 296 mm pitch circle diameter.

The condenser is a Bitzer type VW and has a capacity of 16/8 kW when the logarithmic mean temperature difference is 10°C . The main body of the condenser consists of a horizontal tube, 150 mm diam, 1 m long, which is water cooled. The water passes through plain tubes in the bottom half and then finned tubes in the top half. The bottom half acts as a liquid receiver, and sub-cooling of the liquid refrigerant takes place in the bottom plain tubes whereas condensation takes place on the finned tubes. The oil separator and hot gas bypass valve are situated close to the compressor.

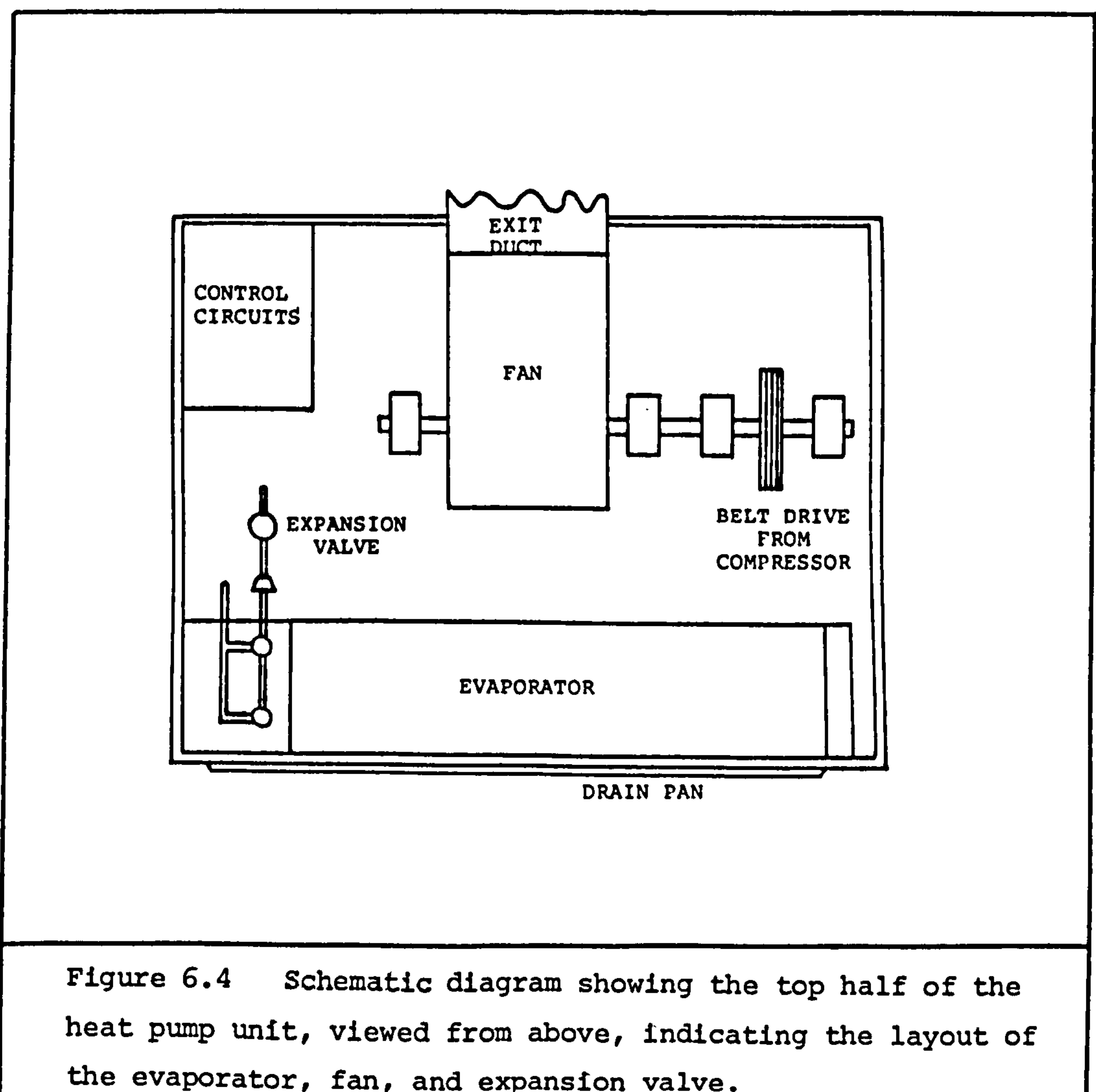
The exhaust gas heat exchanger, formed of 4 $1/2$ -5m of concentric tubing of 1" and 2" diameters, is clipped to the side of the framework. The inner tube is of copper and the outer of mild steel.

6.2.3 Technical Description of Evaporator Assembly

(see Figure 6.4 and Plates III and IV)

The evaporator assembly is also constructed from square sectioned tubing but enclosed by sheet metal panelling to direct the airflow. The overall dimensions of the evaporator, supplied by Searle, are 997mm high x 889mm wide x 202mm

deep, and is of horizontal tube construction fitted with aluminium fins spaced at 8 to the inch. It is divided into two sections, each of 4 rows deep, with the refrigerant flow divided into two parallel paths through each. A drain pan at the base of the evaporator collects condensation and any water resulting from defrost. An extension to the air outlet duct was fitted so that the cold air could be recirculated, thus enabling a range of lower temperatures to be reached in the laboratory. Theoretically the extraction rate is 5.3 kW when the airflow is $0.8 \text{ m}^3 \text{ s}^{-1}$, the inlet temperature is 3°C (dry bulb) and the evaporating temperature is -5°C .



The expansion valve used is an Alco thermostatic valve fitted with an external pressure equalisation connection (Type TCE 4FW). The air fan was supplied by Airscrew Holden Ltd., and is a double entry centrifugal type 102H2WL, of 235 mm diameter. $0.9 \text{ m}^3 \text{ s}^{-1}$ of air can be delivered against a 100 Pa static pressure and the fan absorbs 0.34 kW at 900 rpm.

6.3 Controls

Figure 6.5 overleaf is a diagram of the control circuitry used.

6.3.1 Starting Procedure

Starting of the unit is achieved by a standard Lucas car starter motor, the starter pinion being engaged by the solenoid for as long as the motor is energised. Originally an inertia starter was used, but proved to be unsatisfactory since an initial firing would throw the pinion out from the engine gearing, and the engine often refused to fire on the next compression. This was partly due to having a single cylinder engine, and partly due to stiction in the carburettor at very low gas flow rates. The present starter motor gives reliable automatic starting and is only de-energised when a preset engine speed is sensed. The original design used a centrifugal clutch to offload the engine when starting, however this proved unsatisfactory due to problems of slip, noise and excessive vibration, and was therefore replaced by an electromagnetic clutch, which has proved to be very successful.

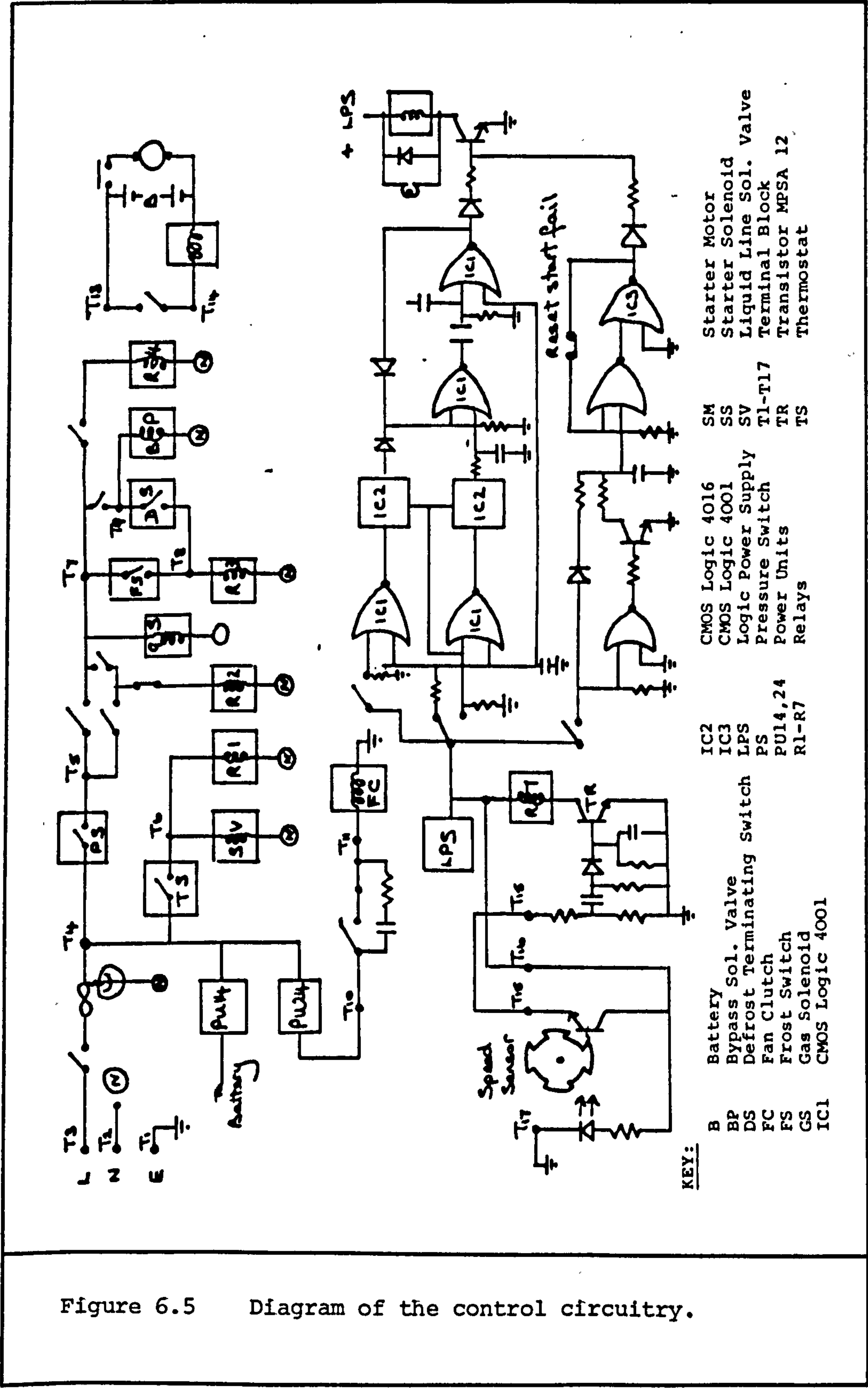


Figure 6.5 Diagram of the control circuitry.

Starting is achieved as follows:-

A 240 volt signal is received and the liquid line solenoid valve opens. Pressure rises in the evaporator, and the low pressure switch closes. The gas solenoid valve opens and gas is admitted to the engine. The heavy current starter relay closes and the starter motor is connected to the battery and starts the engine. The speed sensor causes the starter motor to disengage and stop when a preset speed is achieved. The clutch is engaged manually.

6.3.2 Stopping Procedure

On shutting down the freon is sucked down to the condenser/receiver where it remains until the machine is restarted.

When the 240v signal ceases, the liquid line solenoid valve closes and the compressor pumps the remaining liquid refrigerant out of the evaporator, thereby lowering the pressure until the low pressure switch opens. The gas solenoid valve then closes off the gas supply and the engine stops.

6.3.3 Protection of System

The heat pump has the following safety devices built in:-

(a) The heat pump is turned off by the low pressure switch activating and this also prevents any unwanted starting of

the machine.

(b) If there were a loss of refrigerant, blockage of the expansion valve or any incident causing an excessively low pressure, the low pressure switch again would open and the machine would stop as in the stopping procedure.

(c) If there were, for example, a reduction in water flow through the condenser, or any incident causing an excessively high pressure on the discharge side of the compressor, the high pressure switch opens and the machine would stop as in the stopping procedure.

(d) If the engine does not start satisfactorily within a certain predetermined time, a time delay switch opens and switches off the starter circuit.

6.4 Commissioning of the Heat Pump

6.4.1 Description of Instrumentation, problems encountered, and modifications made

The schematic (Figure 6.6 overleaf) shows the layout of the system with the various monitoring points marked.

All temperatures are measured using platinum resistance thermometer (PRT's)⁺ and are necessary for the basic thermodynamic analysis of the system.

+ Later work has since shown the PRT's suffer long term drift problems and many researchers have reverted to using thermocouples, however at the time the work was carried out we had no cause to doubt our readings. With retrospect the use of PRT's plus the errors associated with averaging of results may well have led to some anomalies in the results quoted in the next Chapter.

T1 to T5 Water temperatures

T7 and T8 Air temperatures

T9 to T12 Freon temperatures

T13 Exhaust temperature

T6 was originally used to measure the temperature of the engine oil, however, this channel was sacrificed in order that water flow could be measured.

Es - Engine speed. This is measured by an optical interrupter on the drive shaft.

Tq - Torque. This is measured by means of a strain gauge bridge on the drive shaft. The signal is amplified on the shaft and feeds an indicator unit via R.F. telemetry.

Torque, together with engine speed gives a direct measurement of the power from the engine which is essential for determining engine efficiency, engine radiation losses, and compressor efficiency.

P1,P2 - Freon pressures. These are measured directly by Schaevitz electronic pressure transducers.

P3 - Inlet gas pressure, measured by a water manometer.

F1 - Gas flow. This is measured by a calibrated domestic gas meter fitted with an optical interrupter switch.

F2 - Water flow. This is measured using a Kent Instrument positive displacement water meter fitted with a reed switch which produces a pulse on a chart recorder for each litre of

water used, or can be sent directly to the data logger.

F3 - Freon flow, measured by an LM221 litre meter.

Freon flow is required for the thermodynamic analysis of the system. How the flow fluctuates in response to the action of the thermostatic expansion valve is of particular interest since good flow control is essential for efficient evaporator heat exchange.

Originally a data unit was built by the Open University's electronics common facilities laboratory, to perform signal conditioning for the PRT's, engine speed, and litre meter channels, having both voltmeter and data logger outputs. Signals from the water meter, gas meter and torque meter indicator unit were passed to a two channel, plus two event marker, chart recorder. However, anomalies in the results occurred, which showed the data unit to be at fault and experiencing a high degree of drift. The problem was partly relieved by modification and frequent recalibration, however a more satisfactory result was achieved by installing a Microdata M1600L 20 channel data logger, borrowed from another OUERG project. The signals from the temperature probes, plus gas, freon, and water flow, freon pressures, and engine speed and torque were passed directly to the data logger and recorded onto cassette tape. The cassette tape could then be read directly onto a teletype, or fed into a computer for further analysis. The data logger contains PRT signal conditioning cards which enable measurement of temperature readings accurately to 0.1°C free of drift.

6.4.2 Initial Running - problems encountered and modifications made

The unit was delivered disassembled from Lucas Aerospace to the Open University on 25th May 1978. Previous to this the machine had been run, but no measurements or development work undertaken. In retrospect, allowing no time for testing and debugging in the programme was a mistake, since this period proved to be very time consuming. The unit was rebuilt, recharged, checked for leaks, and housed in a specially built acoustic enclosure. Initial test facilities were prepared and preliminary instrumentation fitted by October 1978.

The first problem encountered was that of excessive vibration, and was eventually traced to the centrifugal clutch on the engine drive shaft, which was experiencing excessive vibration and slip. On examination the problem was found to be due to a detail design fault. Two solutions were possible: redesign of the drive shaft and clutch, or replacement of the clutch by a solid drive shaft, requiring the engine to start on load. The latter solution was adopted, starting being achieved by operation of the freon bypass valve to off-load the compressor. Following this, in January 1979, the torque instrumentation was received from Astech Electronics. After calibration, direct measurement of engine efficiency and power were possible and efficiencies of only 11% were obtained, instead of the anticipated 18%. A long period then followed, in which carburettor modifications, different carburettor types, gas pressure

variation, and different ignition timings were all evaluated to try and improve the engine efficiency. During May 1979, we totally stripped the engine down to ascertain its general condition, check for valve wear etc, and discovered that the engine had been delivered from the Brit factory with its valve timing 20° out, the root cause of our poor efficiency. With valve and ignition timing set correctly, and with an unmodified CA50 carburettor the efficiency rose to an acceptable 17%. However, starting on load now proved to be a problem, due partly to poor carburation at low speeds where the mixture appeared to be too rich. The single cylinder engine used is particularly prone to this difficulty, having an unsteady gas demand. A better gas regulator, fitted in June 1979 eased the problem, but did not totally overcome it, and starting was still unreliable. To overcome this difficulty, an electromagnetic clutch was fitted in November 1979, along with gas surge tanks to regulate the gas flow.

The electromagnetic clutch allowed the heat pump to start off load, and also had the advantage that by de-clutching at various times during running the torquemeter could be checked and re-zeroed, since it was prone to drift problems.

6.5 Test Programme

6.5.1 Description of Performance Tests

The aim of these tests was to see how the heat pump performed during steady running conditions, under a variety of different operating states. When the machine had stabilised,

and steady running conditions prevailed, the fully instrumented heat pump (see section 6.4.1) was monitored for several hours at a time over a range of evaporating temperatures from -15°C to $+10^{\circ}\text{C}$, for different sets of water temperatures. The sets of water temperatures used were $35-55^{\circ}\text{C}$, $40-60^{\circ}\text{C}$, $45-65^{\circ}\text{C}$, $50-70^{\circ}\text{C}$, and $35-60^{\circ}\text{C}$. The heat pump was designed to run with the $35-55^{\circ}\text{C}$ set of temperatures in mind, for a typical domestic situation. 55°C was envisaged as the lowest practicable output temperature for domestic space heating.

The twelve temperature readings, plus engine speed, torque, freon pressures, gas pressure, and water, gas and freon flows, were recorded at convenient time intervals (normally every 20 secs) onto cassette tapes, and can then be fed into a computer. A preliminary analysis was then carried out immediately, (as the example in Figure 6.7 shows). The computer prints out the readings obtained, followed by a brief system analysis based on the following calculations:-

Input power = $1077 \times \text{gas flow (kW)}$ where 1077 is the energy content of 1 ft^3 of gas, after correction for the calibration of the gas meter, i.e.

$$1 \text{ ft}^3 = \frac{1017 \times 1.0551}{.9965} = 1077 \text{ kJ}$$

Output power = $\frac{\text{water flow}}{0.993} \times 4.1868 (T_5 - T_1) \text{ (kW)}$, where .993 is a correction for the calibration of the water meter and 4.1868 is the specific heat capacity of water ($\text{Jkg}^{-1}\text{C}^{-1}$). T_1 is the temperature of water entering the system,

- Heat pump test run Time 4286 seconds File date

SYSTEM DATA

```

-----
Input water temperature   T1  = 35.8  C
Condenser water input     T2  = 35.8  C
Condenser water output    T3  = 45.1  C
Jacket water temperature  T4  = 58.6  C
Output water temperature  T5  = 55.4  C

Air input temperature     T7  = 3.8  C
Air output temperature    T8  = 3.8  C

Evaporator freon in       T9  = 7.3  C
Evaporator freon out      T10 = 8.8  C
Condenser freon in        T11 = 66.3  C
Condenser freon out       T12 = 48.1  C

Exhaust air out temp      T13 = 38.3  C

Engine speed              = 1831 rpm
Engine torque (corrected) = 571  nv

Condenser pressure        = 9.23 bar
Evaporator pressure       = 2.35 bar

Water flow rate           = 8.155 l/sec
Gas flow rate             = 8.115 cu ft/sec
Freon mass flow rate      = 8.844 kg/sec
  
```

SYSTEM ANALYSIS

```

-----
INPUT POWER              11.47kW
OUTPUT POWER             13.33kW
COEFFICIENT OF PERFORMANCE 1.162

Output breakdown
Exhaust condenser        8.52kW
Freon condenser          6.88kW
Jacket                    3.59kW
Exhaust                   3.14kW

CONDENSER
Saturation temperature   38.34C
Superheat temperature    27.96C
Enthalpy h input         224.15
                        h output    74.68
                        h difference 149.47
Freonflow Litre/minute  8.841 l/s
Litreneter               8.844

EVAPORATOR
Saturation temperature   -7.77C
Superheat temperature    5.87C
Enthalpy h output        187.85
                        h difference 113.17
Extraction               4.68kW

COMPRESSOR
Compressor speed         867.8
Fan power                 8.372
Engine output power      1.82
Compressor power         1.36
Manufacturers power available 1.88 kW

ENGINE
Gas input 11.47kW
Power out 1.82 kW 15.98%
Exhaust 3.66 kW 31.91%
Jacket 3.59 kW 31.34%
Losses 2.39 kW 28.86%
  
```

Figure 6.7 Example of type of Computer Print Out enabling spot checks of the system to be carried out. In this particular print out T10, the temperature of the freon leaving the evaporator was not functioning correctly.

and T5 the water leaving the system.

$$\text{C.O.P. Co-efficient of Performance} = \frac{\text{Output Power}}{\text{Input Power}}$$

The output power (in kW) can then be broken down further, into individual components.

Exhaust condenser	=	$(T_2 - T_1)Q$	
Freon condenser	=	$(T_3 - T_2)Q$	
Jacket	=	$(T_4 - T_3)Q$	
Exhaust	=	$(T_5 - T_4)Q$	

Where Q, as above
= $\frac{\text{water flow}}{.993} \times 4.1868$

The various component parts of the system can then be analysed individually as follows:-

Condenser

The pressure reading is measured in volts, however this is converted directly from manufacturers data and printed in bars. The saturation temperature Tsat can then be directly obtained from the refrigerant tables for Freon 12.

Number of degrees superheat present = $T_{11} - T_{sat}$, where T_{11} is the temperature of the freon entering the condenser. The change in enthalpy Δh is the difference in enthalpies into and out of the system, measured in kJ/kg.

h_{in} is again obtained from refrigeration tables, as is h_{out} , given the appropriate superheat value. The freon flow rate is then calculated by

$$\frac{\text{condenser output (kg/sec)}}{\Delta h}$$

This was necessary since the litre meter reading could not be relied upon due to fluctuating flow conditions.

Evaporator

Again the pressure reading is measured in millivolts, and is converted to bars from the manufacturers' data. The saturation temperature Tsat is read from refrigeration tables of pressure v. temperature, and the superheat is given by $T_{10} - T_{sat}$ where T_{10} is the temperature of refrigerant leaving the evaporator.

The change in enthalpy Δh is again given by the difference in enthalpies in and out of the system, however it should be noted here that h_{in} is the same as h_{out} from the condenser. h_{out} can be read from refrigeration tables of T_{sat} v. enthalpy. The heat extraction by the evaporator measured in kW, is then equal to the Freon flow $\times \Delta h$.

Fan

Given the extraction rate, the quantity of air used can be determined

$$\text{Mass of air moved by fan} = \frac{\text{heat extraction (kg/sec)}}{1.005 (T_7 - T_8)} \text{ where}$$

T_7 and T_8 are the temperatures of air into and out of the evaporator respectively, and 1.005 is a constant, the specific heat capacity of air.

The volume of air moved is given by $1.225 \times \text{mass of air}$ where 1.225 is the density of air (kg m^{-3}).

Compressor

Engine output power (kW) = $0.000001746 \times \text{Engine speed (rpm)} \times \text{Torque (mv)}$ where 0.000001746 is a constant k where

$$k = \frac{16.68 \times 2\pi}{60 \times 10^6}$$

used to convert the torque reading in mv to Newton metres ($1 \text{ v} = 16.68 \text{ Nm}$) and the speed to revs per sec.

Compressor speed is given by $\frac{\text{Engine speed}}{2.11}$

where 2.11 is a constant referring to the drive ratio between the engine and compressor.

Fan power is given by $390 \frac{(\text{Engine speed})^3}{1860}$

where 390 is a constant supplied from manufacturers' data. The fan is quoted to have a power consumption of 0.39 kW at 1860 rpm. The fan power is assumed to vary as the cube of speed.

The power to the compressor can be found by subtracting the fan power from the shaft power, i.e. $(\eta \text{ belts} \times \text{engine output power}) - \text{fan power}$ where η is the efficiency of the belt drive, which is normally very efficient, and in this case assumed to be 0.95.

This figure can be checked using the manufacturers' data for the compressor at 1,000 and 5 degrees of superheat. The

output power can be obtained from curves of varying condensing temperatures on a graph of output v. evaporating temperature, and this figure multiplied by the engine speed to give the manufacturers' power available.

Finally, an engine heat balance is carried out and the efficiencies of various parts calculated.

This initial analysis work then forms the basis for further analysis, the results of which are summarized in the next Chapter .

6.5.2 Control Strategy Tests

Whether or not engine driven heat pumps prove to be a viable method of heating or supplying hot water, will mostly depend not upon the efficiency at full load but upon the efficiency of the system at part load, since these are the conditions prevalent for the majority of the heating season. Thus it was felt important to try out different control strategies on the system, apart from that used in the performance tests where the machine ran, as designed, on full throttle to maximize efficiency, and running speed was determined by load.

There are two basic ways of control of engine driven heat pumps:-

(a) running intermittently, or (b) modulation.

At near to peak loads, modulation by engine throttle control would appear to be most efficient; however at very low loads

either the engine efficiency drops drastically or, indeed, the engine cannot physically be run at a low enough throttle setting to ensure continuous operation. At this end of the scale intermittent running would seem to be best. Thus, there is a point, known as the 'crossover load', above which heat is supplied most efficiently by modulating the speed of the heat pump, and below which intermittent running is most efficient.

The aim of these tests, then, were to identify this crossover load for our machine in the laboratory, based on the assumption that a typical building time constant is large compared with a heat pump time constant i.e. our heat pump is able to reach a steady state condition in a very short time compared to that in which a building would be expected to vary. In practice, the heat pump reaches a steady state condition in about 15 minutes. Thus by using our system to supply steady part loads by throttling (modulating) and intermittent running, the crossover point and the factors governing it could be determined. However, due to the characteristics of our particular single cylinder engine, running at anything less than full throttle proved completely impossible, and only intermittency tests could be carried out. Unfortunately these also had to be curtailed prematurely as explained in the introduction on page 12 and the analysis remains far from complete.

Three sets of cycling times were used, 5, 10 and 15 minutes. Using the 5 minute set as an example:-

(i)	5 minutes	h.p.	on	water pump	on)	
	"	"	off	"	")	
	"	"	on	"	")	average C.O.P.
	"	"	off	"	")	obtained for this
	"	"	on	"	")	30 minute run.
	"	"	off	"	")	
(ii)	5 minutes	h.p.	on	water pump	on)	
	"	"	off	"	")	
	"	"	on	"	")	average C.O.P.
	"	"	off	"	")	obtained for this
	"	"	on	"	")	30 minute run.
	"	"	off	"	")	
(iii)	30 minutes	h.p.	on,	water pump	on)	average C.O.P.
							obtained.

The average C.O.P's obtained from each of the 3 thirty minute sampling periods can then be compared.

6.5.3 Defrost Tests and Results

Defrosting an ice build up on the evaporator is achieved by a hot gas bypass, i.e. a solenoid valve opens and allows hot gas from the compressor to return directly to the evaporator, thus melting away any ice. The defrost procedure is initiated when a preset temperature differential between the freon vapour and ambient air is exceeded. The control system de-energises the clutch causing the fan to stop, and the hot gas bypass valve opens. When the freon evaporating temperature exceeds a preset value, the defrost is terminated. Ideally the defrost should operate for just sufficient time for the build up of ice on the evaporator to be cleared, and no longer. Although the theory behind the procedure is simple, it was felt that the system should be

tried out in practice, although not part of the original test programme. To enable ice to form in the laboratory, a fine water spray was used to inject droplets into the recirculating ductwork.

After fine adjustment of preset values initiating and terminating defrost, it was found that the defrost worked very quickly indeed, and on average 3 litres of melted ice could be collected after a defrost burst of 5 minutes.

Although the method of hot gas bypass was proved to be successful, the required frequency of defrost was unable to be ascertained, again due to the project having to be curtailed prematurely.

CHAPTER 7 OUERG/LUCAS HEAT PUMP - RESULTS

This chapter discusses the results obtained from the test programme as described in the previous chapter. The results are analysed to give the performance of each of the component parts of the system in order to assess the overall performance of the gas fuelled engine driven heat pump⁺. In general, the results stated are based on mean values.

7.1 Engine Performance

During the analysis of engine results it became clear from derived and measured values that the engine speed as measured was 5% low, and thus suitable corrections were made to the results. During the testing itself, the mixture strength varied due to inadequate gas control, and the torque readings were unreliable due to temperamental batteries and temperature effects. Some correction, therefore, had to be made, and by plotting torque against thermal efficiency (Figure 7.1), an inverse consumption loop is obtained showing that a maximum efficiency of 17.8% is achieved with a torque of 10.5 Nm, and that the maximum torque of 11.7 Nm. occurred at an engine efficiency of 16.8%. (Tests carried out under full throttle conditions).

+ It should be noted that, at the time the work was carried out, we had full confidence in our instrumentation, however as the footnote on page 114 explains, the use of PRT's may have led to some anomalies in the results.

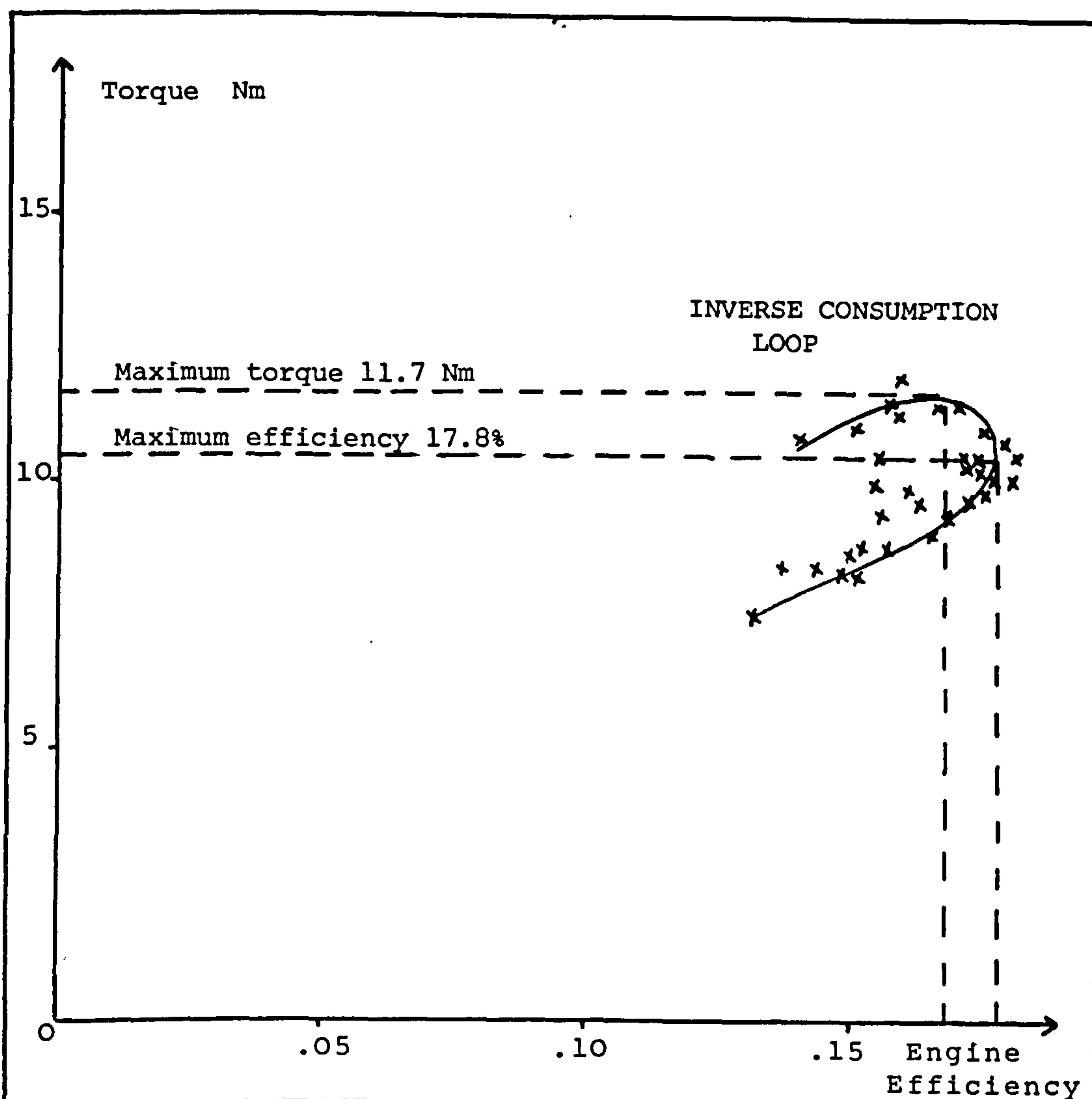


Figure 7.1 Plot of torque versus engine efficiency (measured values), Engine full throttle.

Now a plot of Torque against Relative Mixture Strength can then be made (Figure 7.2) - where

$\text{rel.mix.strength} = \frac{\text{gas flow}}{\text{air flow}}$ with respect to stoichiometric.

$\text{gas flow} = \frac{\text{gas input (kW)}}{1077} \times 2.83 \times 10^{-2} \text{ m}^3 \text{ sec}^{-1}$

$\text{air flow} = 0.366 \times 10^{-3} \times \eta_{\text{vol}} \times \frac{\text{r.p.m.}}{60 \times 2} \text{ m}^3 \text{ sec}^{-1}$
(cylinder capacity) (4 stroke engine)

The volumetric efficiency η_{vol} was discovered to be 0.66, rather than the optimistic figure of 0.85 used in the design

calculations.

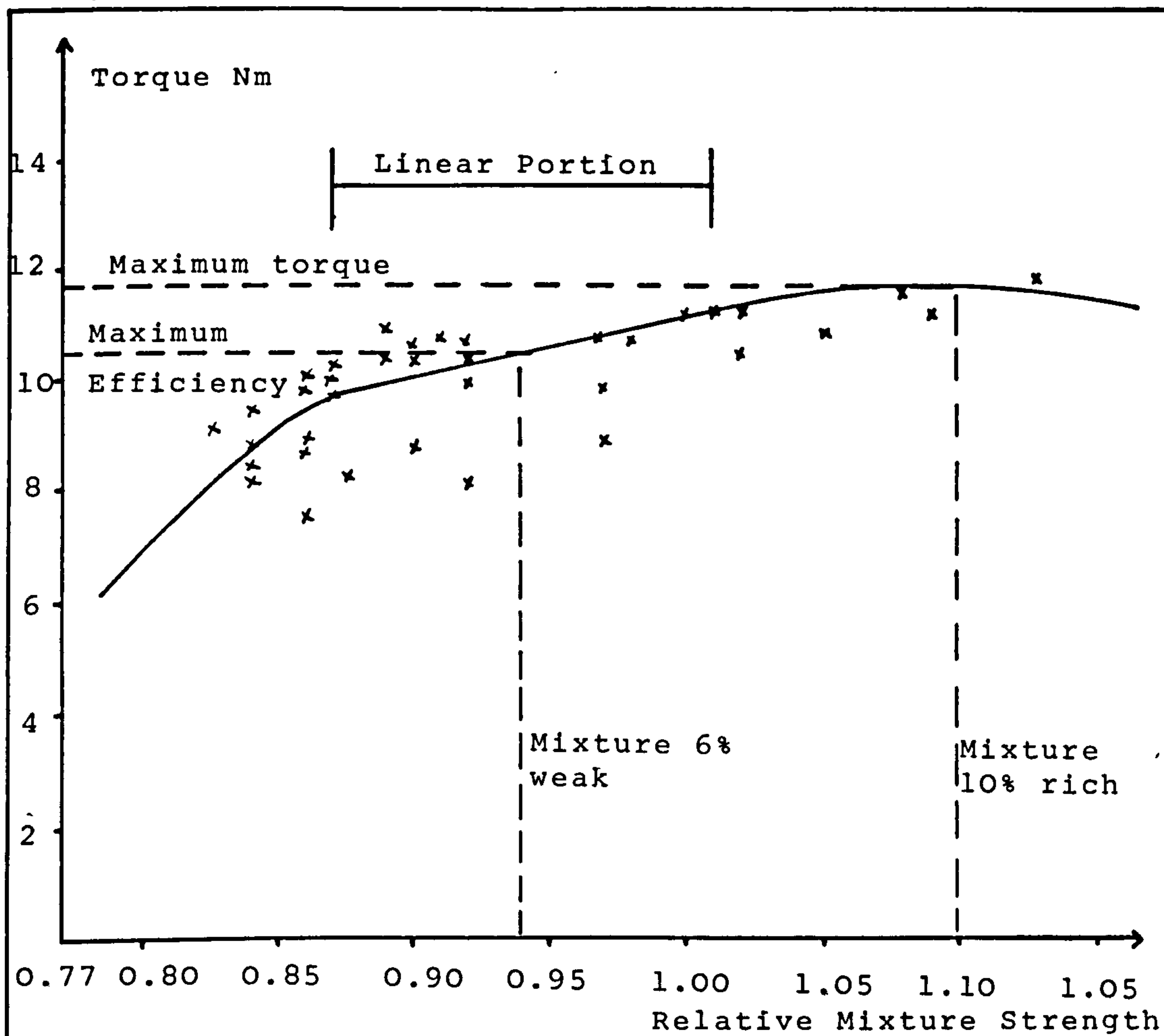


Figure 7.2 Plot of torque versus relative Mixture strength.

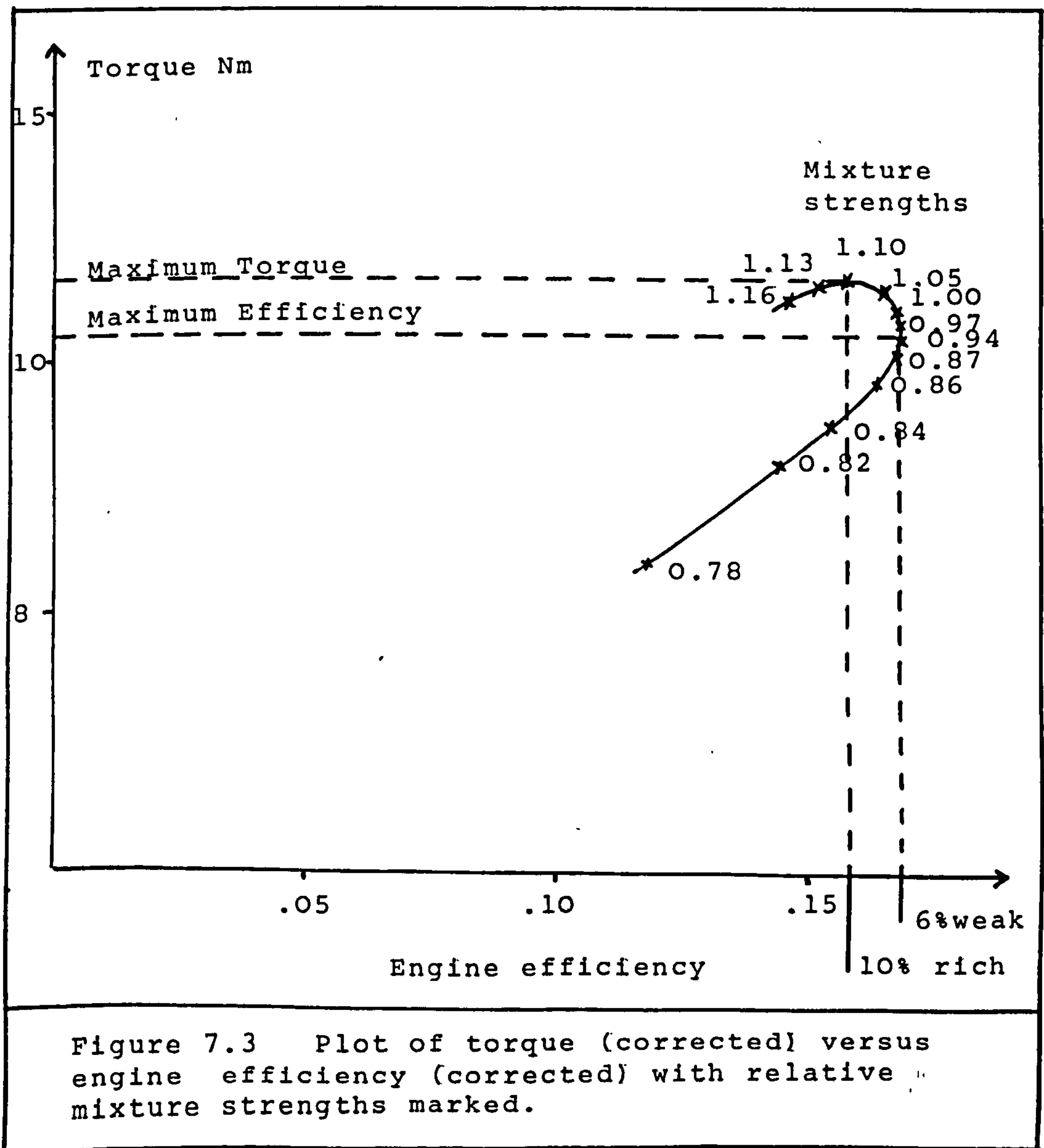
From the analysis of the results the 'best' curve is taken as linear between mixture strengths 0.87 and 1.01 and is continued by curves at both ends to fit the remaining points.

As the mixture strength increases richer than stoichiometric, it reaches a maximum torque value when 10% rich. The torque then falls off with further richening of the mixture. The torque also falls off faster below 0.87 stoichiometric, due probably to incomplete combustion of this lean mixture.

Over the linear portion of the graph, the torque increases

with increasing mixture strength, i.e. increased fuel means increased combustion temperature, therefore increased cylinder pressure.

Thus, from the original calculations of mixture strength, new corrected values for torque can be read off from the curve, and new power values can be calculated. From these values a new efficiency (shaft power/gas input power) can be calculated, leading to a new plot of Torque against efficiency (Figure 7.3). This figure shows that a maximum efficiency of 16.9% was obtained with a mixture strength of 6% weaker than stoichiometric, giving a torque of 10.5 Nm.



The maximum torque of 11.7 Nm occurred with a mixture strength of 10% richer than stoichiometric, however the efficiency then dropped to 15.8%. Table 7.1 shows the difference between design performance figures and those actually achieved in practice.

Table 7.1 Engine Performance Figures for Design Point

	Design	Actual
Engine Speed (rpm)	1860	1860
Mixture Strength	10% weak	6% weak
Volumetric Efficiency η_{vol} (%)	0.85	0.66
Gas Input (kW)	15.08	12.08
Thermal Efficiency η (%)	18.50	16.90
Shaft Power (kW)	2.79	2.04
Torque (Nm)	14.3	10.5

It should be noted here that if the carburettor had been operating correctly, i.e. a constant mixture strength, no such drop in efficiency as seen in Figure 7.3 would have existed. Therefore, all the tables are derived from these results, assuming a constant air/fuel ratio mixture strength.

Table 7.2 overleaf shows the engine performance at different speeds assuming a constant thermal efficiency of 16.9%. The engine shaft power then varies proportionally with speed, as can be seen in Figure 7.4.

Table 7.2 Engine Performance at Different Speeds.

Engine Speed rpm	1200	1400	1600	1800	1860	2000
Shaft Power kW	1.32	1.54	1.76	1.98	2.05	2.20
Efficiency % gross Cv	16.9	16.9	16.9	16.9	16.9	16.9
Cooling water heat rate kW	2.51	2.93	3.35	3.76	3.89	4.18
Reclamation from exhaust kW	2.35	2.74	3.13	3.52	3.64	3.91
Losses kW	1.65	1.93	2.20	2.48	2.56	2.75
Fan power kW	0.13	0.18	0.24	0.32	0.37	0.41
Power to Compressor kW	1.19	1.36	1.52	1.66	1.68	1.79

From the results obtained it can be shown that the cooling water reclamation (with water in at 35°C and out at 55°C) was 4% for the condenser and 27% for the heat exchanger, i.e. total of 31%. Radiation and other losses = 100 - 80 = 20%.

The fan power is assumed to vary as the cube of the speed; at each engine speed the fan power has been calculated and subtracted from the shaft power to give the power available for driving the compressor, as shown in Figure 7.4.

7.2 Condenser Performance

An analysis of all the test results showed that the manufacturers' claim that the condenser has a capacity of removing 1.68 kW of heat per degree of logarithmic mean temperature difference was accurate.

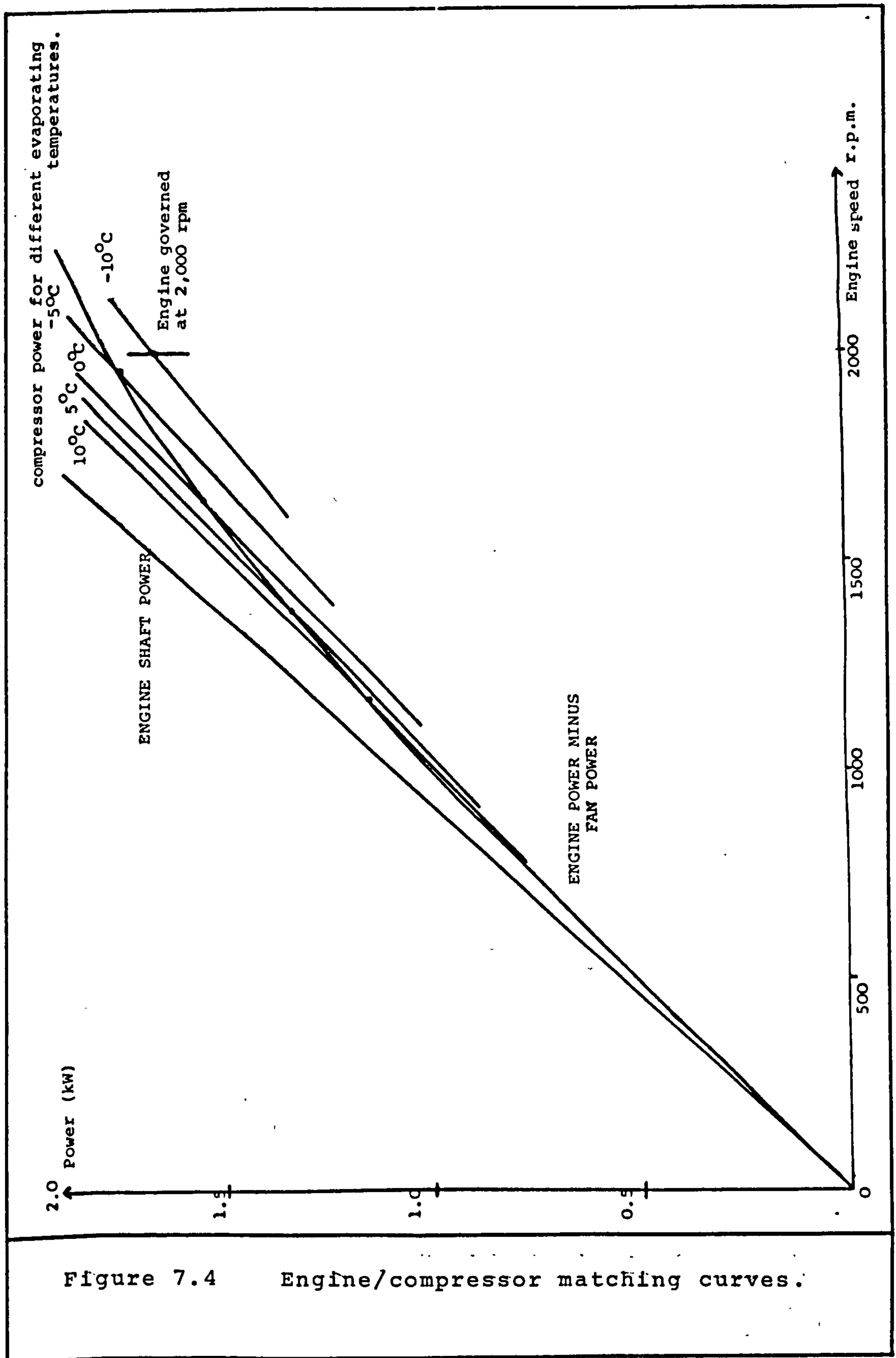


Figure 7.4 Engine/compressor matching curves.

Table 7.3 Condenser Performance

Log Mean Temp. Diff. °C	Heat Capacity kW	Condensing Temperature °C
1	1.7	45.8
2	3.4	45.9
3	5.0	46.3
4	6.7	46.8
5	8.4	47.5
6	10.1	48.4
7	11.8	49.2
8	13.4	50.1
9	15.1	51.0
10	16.8	51.9

The condenser power $P = k \Delta t_m$ where k is the heat transfer constant (1.68 from manufacturers' data) and

$$\Delta t_m = \frac{\Delta t_1 - \Delta t_2}{\ln \Delta t_1 / \Delta t_2} = \text{log mean temperature difference}$$

$$\begin{aligned} \text{now } \Delta t_1 &= T_c - T_i \quad \text{and} \quad T_c = \text{condensing temperature} \\ \Delta t_2 &= T_c - T_o \quad \quad T_i = \text{water in temperature} \\ &\quad \quad \quad T_o = \text{water out temperature} \end{aligned}$$

Subcooling is given by $T_c - T_{12}$ where T_{12} is the refrigerant temperature leaving the condenser. During the tests the subcooling varied between the values of 4.4 and 7.2°C.

7.3 Compressor Performances

The results of the tests for the compressor show that the measured extraction rate was some 81% of that claimed by the manufacturers. There are three possible reasons for this

discrepancy.

- (1) The manufacturers' data has 5% tolerance;
- (2) The manufacturer works on a suction temperature of 18°C, whereas this could be higher in tests due to the temperature inside the enclosure;
- (3) The manufacturer calculates extraction using the evaporator temperature and pressure, and the pressure drop in the pipe between the evaporator and compressor suction could, therefore, have a measurable effect.

Table 7.4 Derived Compressor Performance for Various Evaporator Temperatures, and driven at 1000rpm.

Evaporating Temps. °C	-10	-5	0	5	10
Shaft Power kW	1.75	1.87	1.97	2.04	2.09
Extraction Rate kW	3.66	4.69	5.88	7.28	8.80

The measured compressor power was 78% of that calculated from the manufacturers' data, and again the same factors as mentioned above would come into force.

During development the belt drive ratio was changed from 1.86/1 to 2.11/1 since the engine developed less power than expected. The ratio from the compressor to the fan is 1.1/1. The power available to the compressor, Pa, at 900 rpm is given by:

= 0.95 Pe -

1.1

x

340

x

$n^3 / (2.11 \times 1.1 \times 900)^3$

main drive loss

fan drive loss

340 watts

drive ratios

engine/
compressor

compressor/
fan

where P_a is the engine shaft power = $1.1 n$ where n is the engine speed in rpm.

The expression can be simplified to read:

$$P_a = 1.045 n - 4.1 \times 10^{-8} n^3.$$

The compressor power required, P_c , for condensing at 47.5°C , is given by

$P_c = \text{a constant } kX$ where

X is the compressor speed = $n/2.11$.

P_c is given by a fraction of the manufacturers' data 0.78, where 0.78 is the ratio of the mean power for each test, to the manufacturers' power for that test.

Thus for evaporating temperatures:

	k	$k/2.11$
- 10 °C	1.75	0.829
- 5 °C	1.87	0.997
0 °C	1.97	0.934
5 °C	2.04	0.967
10 °C	2.09	0.990

However, under actual operating conditions, $P_a = P_c$, giving the intersection of the compressor extraction lines and the engine power available curve.

$$1.045 n - 4.1 \times 10^{-8} n^3 = \frac{k}{2.11} n$$

$$n = \frac{(1.045 - k/2.11)}{4.1 \times 10^{-8}}$$

Therefore, the points of intersection where a power balance exists between the engine and the compressor, illustrated in Figure 7.4 are shown in Table 7.5.

Table 7.5 Engine / Compressor Matching Points

Evaporating Temp. °C	Engine Speed rpm	Power kW
-10	2,294	1.9 (governed to 1.66kW at 2000 rpm)
-5	1,962	
0	1,645	1.54
5	1,379	1.33
10	1,158	1.15

The manufacturers' data is based on suction saturated temperature and 18°C actual entry temperature, whereas the evaporating temperature used may be higher than this suction saturated temperature because of a pressure drop in the suction piping. Also, the inlet temperature could be around 25°C rather than 18°C. The compressor shaft power and extraction rate are shown in Table 7.6 and also in Figure 7.5, versus evaporating temperature.

Table 7.6 Compressor Performance under Various Evaporating Temperatures.

Evaporating Temp. °C	-10	-5	0	5	10
Compressor Speed rpm	948*	930	780	654	549
Compressor Shaft Power kW	1.86	1.74	1.54	1.33	1.15
Extraction Rate kW	3.47	4.36	4.59	4.76	4.83
Heat to Condenser kW	5.13	6.10	6.13	6.09	5.98
Total Engine Power kW	2.20	2.16	1.81	1.52	1.27
Heat Pump C.O.P.	2.33	2.82	3.39	4.01	4.71

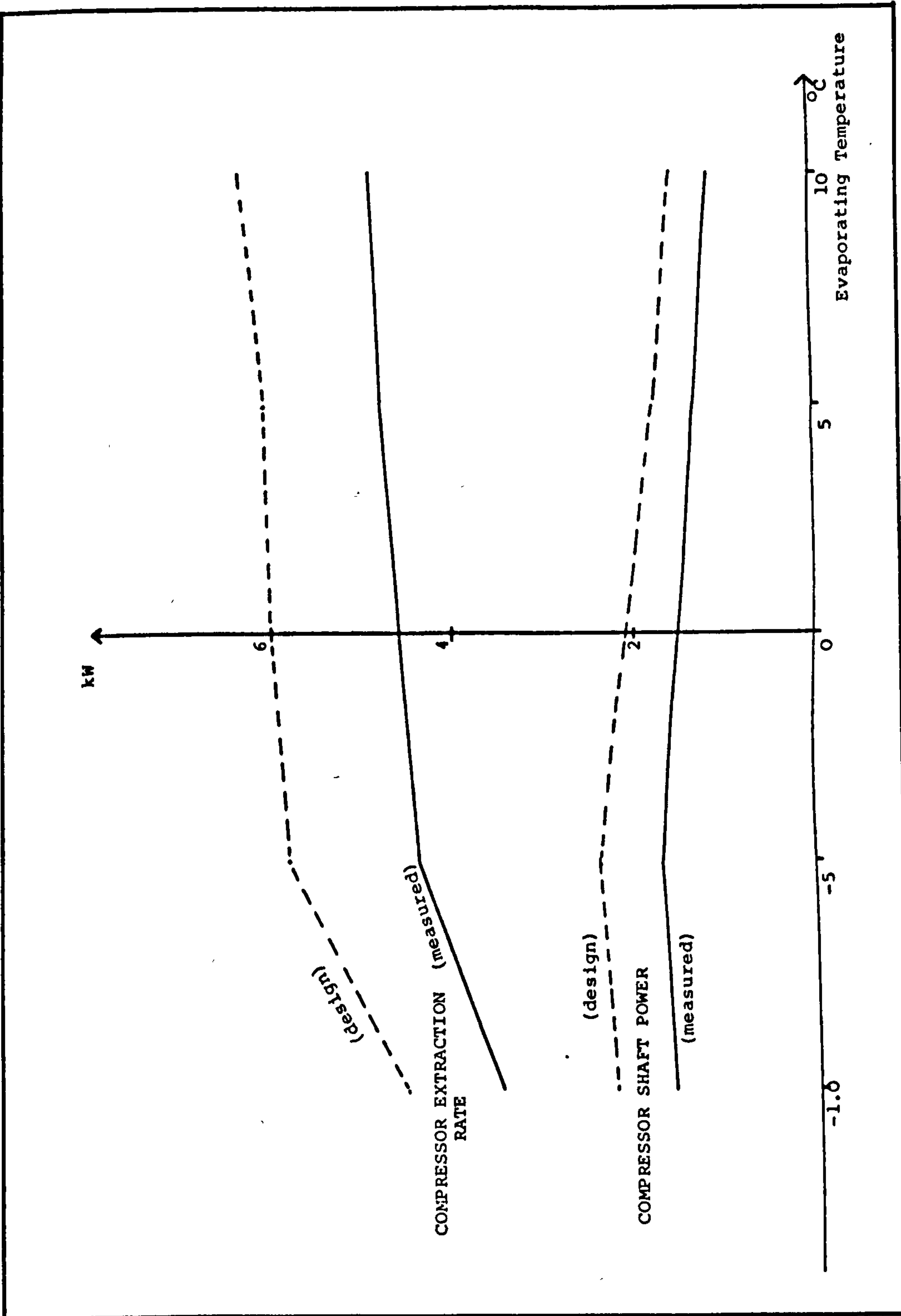


Figure 7.5 Plot of compressor extraction and power versus evaporating temperature.

The Coefficient of Performance,

$$\text{C.O.P.} = \frac{\text{heat pump output (heat to condenser)}}{\text{total engine power}}$$

The C.O.P. is shown, along with the heat pump output, plotted against evaporating temperature, in Figure 7.6 and against ambient temperature in Figure 7.7.

7.4 Evaporator Performance

The evaporator design point was taken to be for 3°C ambient, an evaporating temperature T_e of -5°C, operating at an engine speed of 1,860 rpm. However, the measurements obtained indicate that when operating at an evaporating temperature of -5°C, the ambient temperature is 5.8°C at an engine speed of 1,962 rpm.

A graph of the results obtained by plotting evaporator air in temperature (ambient) against the evaporating temperature can be seen in Figure 7.8, where the relationship can be given in the form

$$T_i = 1.386 T_e + 12.76 \text{ where}$$

T_i is the temperature of the air entering the evaporator, i.e. ambient temperature.

Table 7.7 on page 145 shows the Evaporator Performance related to ambient temperatures.

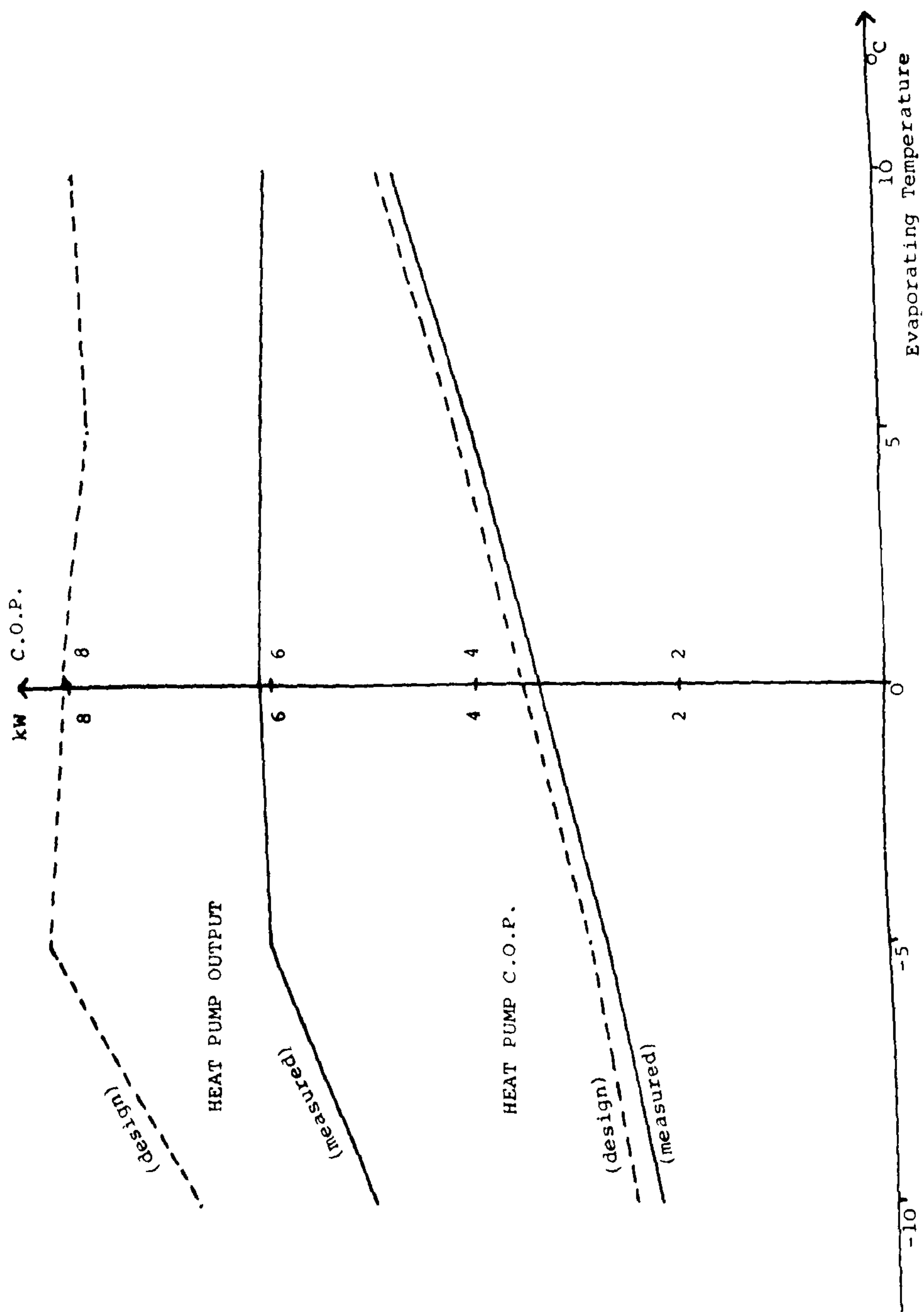


Figure 7.6 Plot of heat pump output and C.O.P. versus evaporating temperature

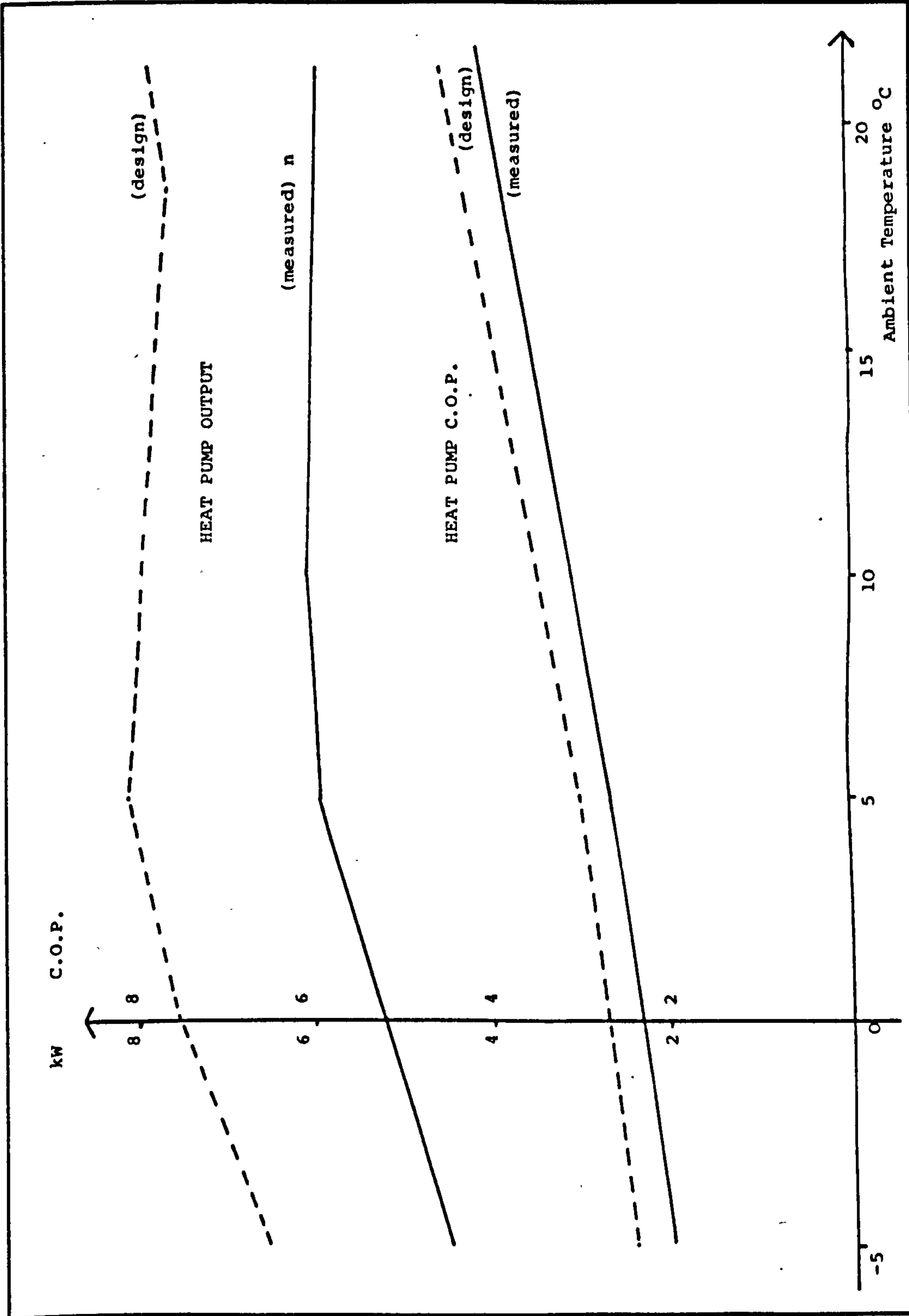


Figure 7.7 Plot of heat pump output and C.O.P. versus ambient temperature.

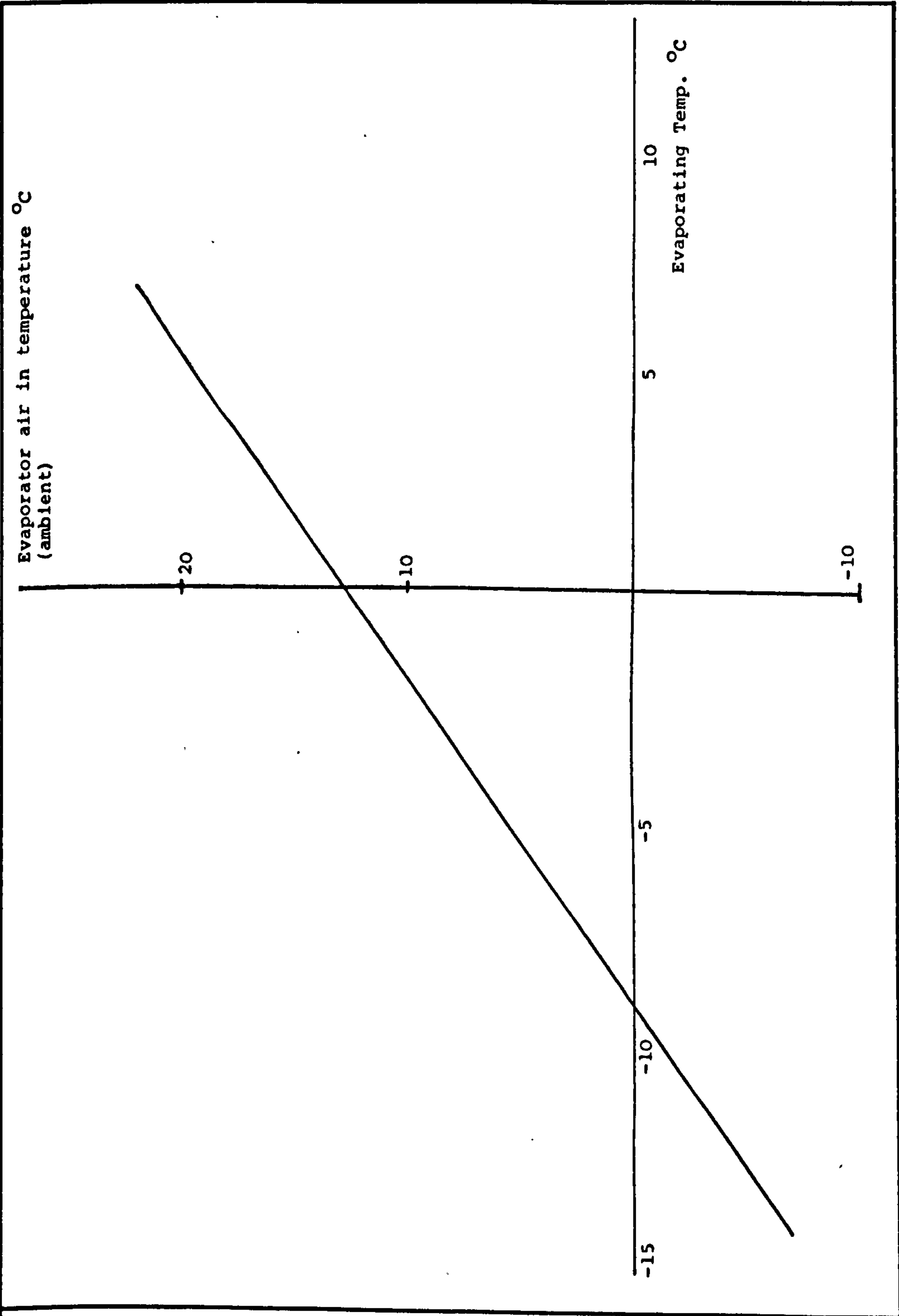


Figure 7.8 Evaporator air inlet temperature (ambient) versus evaporating temperature.

Table 7.7 Evaporator Performance Figures.

Evaporating Temp. °C	-10	-5	0	5	10
Compressor Extraction Rate kW	3.47	4.36	4.59	4.76	4.83
Air Flow m ³ sec ⁻¹	0.86	0.85	0.71	0.59	0.50
Log Mean Temp. Difference °C	7.1	8.5	9.9	11.3	12.7
Air In °C	-1.1	5.83	12.76	19.69	26.62
Air Out °C	-4.5	1.5	7.5	13.5	19.5

7.5 Engine Energy Balance

Table 7.8 gives the results (actual and predicted) of a breakdown of the power consumed by the engine at the design running condition.

Table 7.8 Engine Energy Flows and their Percentage of Energy Input.

	Predicted		Actual	
	kW	%	kW	%
Shaft Power	2.79	18.5	2.05	17
Jacket Cooling Water	4.52	30.0	3.88	32
Main Exhaust Gas Heat Exchanger	3.85	25.5	3.28	27
Second Exhaust Gas Heat Exchanger	1.57	10.4	0.36	3
Heat Lost in Exhaust Gas	0.48	3.2	0.36	3
Heat Lost by Radiation etc.	1.87	12.4	2.20	18
Total	15.08	100	12.13	100

Thus, of the 12.06 kW gas input power to the engine, 17% benefits from the heat pump C.O.P., 62% is reclaimed, and 21% is lost by the residual heat content in the exhaust gas and engine radiation.

7.6 Heat Reclamation from Engine

The design water temperatures of the recirculating water were taken to be 35°C and 55°C, which would have been the temperatures from and to a house heating system in a real situation. These, then, were the temperatures used in most of the tests, although other temperatures were used to see what effect they made to the performance. Table 7.9 shows the change in water temperature as it flows through the system, given the design water temperature. The values of heat output are taken from Tables 7.6 and 7.8 for an evaporating temperature of -5°C.

Table 7.9 Changes in Water Temperature Through the System.

Component	Heat Output kW	Water Temperature °C	
		Rise	Discharge
Second Exhaust Heat Exchanger	0.36	0.53	35.53
Condenser	6.10	8.96	44.49
Engine Water Jacket	3.88	5.70	50.19
Main Exhaust Heat Exchanger	3.28	4.82	55.01

7.7 Total Heat Output and Overall Efficiency

Table 7.10 on page 148 is a summary of the relationships between evaporating temperature, ambient temperature, power inputs and heat flow rates to the total output and efficiency of the heat pump system, and is shown plotted in Figure 7.9. overleaf.

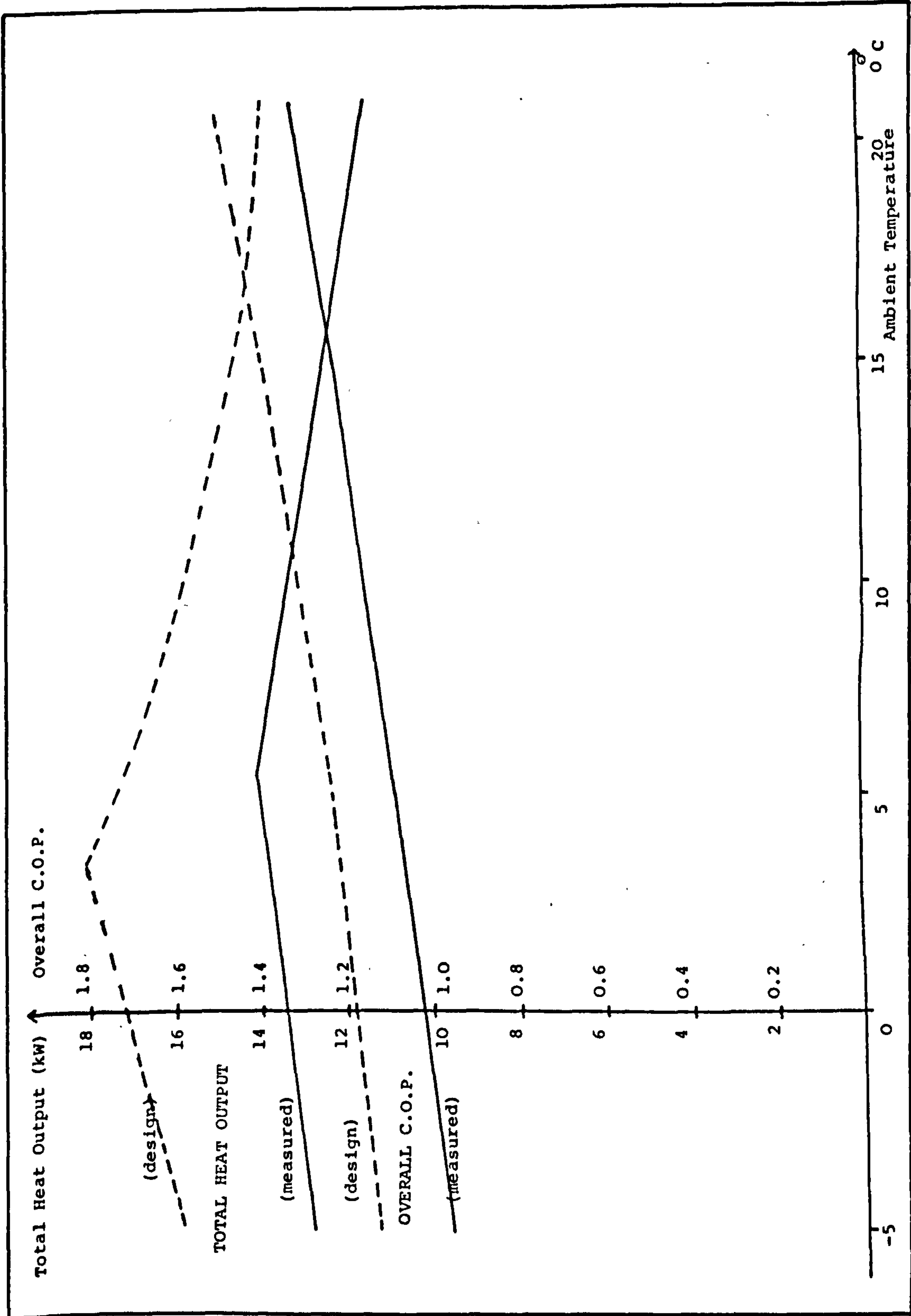


Figure 7.9 Plot of total heat output and overall efficiency versus ambient temperature.

Table 7.10 Total Heat Output and Overall Efficiency.

Ambient Temperature °C	-1.1	5.83	12.76	19.69
Evaporating Temperature kW	-10	-5	0	5
Engine Shaft Power kW	2.20	2.16	1.81	1.52
Gross Gas Input Power kW	13.02	12.78	10.71	8.99
Heat Pump Output kW	5.13	6.10	6.13	6.09
Heat Reclaimed kW	8.07	7.92	6.64	5.57
Total Heat Output kW	13.20	14.02	12.77	11.66
Overall C.O.P.	1.01	1.10	1.19	1.30

Figure 7.10 shows the relationship between ambient temperature and the water and air temperatures within the heat pump.

7.8 Heat Pump Performance Under Varying Water Temperatures

As explained earlier, (Section 6.5), the performance tests were carried out for a range of inlet and outlet water temperatures. So far in the analysis we have confined ourselves to the set used for design purposes. Tables 7.11 and 7.12 give the results for the other sets of water temperatures used, and Figure 7.11 is a plot of C.O.P. against evaporating temperature for the different sets.

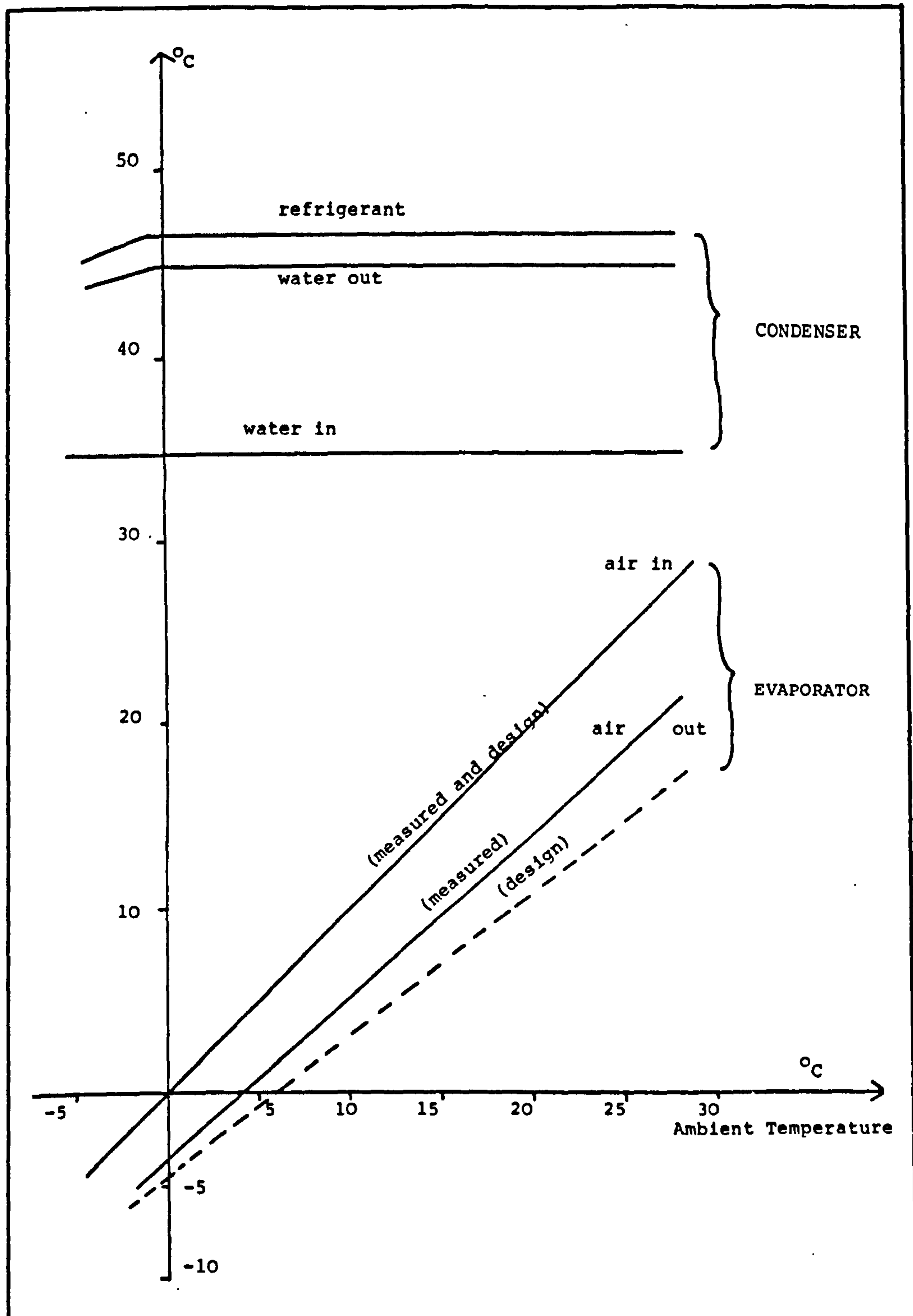


Figure 7.10 Plot of heat pump water and air temperatures versus ambient temperature.

Table 7.11 Compressor Performance at 1000 rpm for Different Condensing Temperatures.

Group *	Evaporating Temp. °C	-10	-5	0	5	10
I (35-55°C)	Condensing Temp. °C	47.5	47.5	47.5	47.5	47.5
	Shaft Power kW	1.75	1.87	1.97	2.04	2.09
	Extraction Rate kW	3.66	4.69	5.88	7.28	8.80
II (40-60°C)	Condensing Temp. °C	52.5	52.5	52.5	52.5	52.5
	Shaft Power kW	1.81	1.95	2.07	2.18	2.25
	Extraction Rate kW	3.35	4.35	5.47	6.82	8.47
III (35-60°C)	Condensing Temp. °C	50.0	50.0	50.0	50.0	50.0
	Shaft Power kW	1.77	1.90	2.00	2.09	2.16
	Extraction Rate kW	3.54	4.55	5.66	7.10	8.59
IV (45-65°C)	Condensing Temp. °C	57.5	57.5	57.5	57.5	57.5
	Shaft Power kW	1.90	2.06	2.20	2.33	2.43
	Extraction Rate kW	3.04	3.95	5.02	6.31	7.67
V (50-70°C)	Condensing Temp. °C	62.5	62.5	62.5	62.5	62.5
	Shaft Power kW	1.47	2.14	2.29	2.45	2.55
	Extraction Rate kW	2.70	3.58	4.61	6.16	7.19

*N.B. Groups I to V refer to different sets of inlet and outlet water temperatures.

Table 7.12 Heat Pump Performance.

Group	Evaporating Temp. °C	-10	-5	0	5	10
I 35-55 °C	Engine Shaft Power kW	2.20	2.16	1.81	1.52	1.27
	Gross Gas Input Power kW	13.02	12.78	10.71	8.99	7.51
	Heat Pump Output kW	5.13	6.10	6.13	6.09	5.98
	Heat Reclaimed kW	8.20	8.05	6.75	5.66	4.73
	Total Heat Output kW	13.33	14.15	12.88	11.75	10.71
	C.O.P.	1.02	1.11	1.20	1.31	1.43
II 40-60 °C	Engine Shaft Power kW	2.13	1.89	1.37	0.66	Engine Stalled
	Gross Gas Input Power kW	12.60	11.18	8.11	3.91	
	Heat Pump Output kW	4.74	5.13	4.46	2.58	
	Heat Reclaimed kW	7.75	6.88	4.99	2.40	
	Total Heat Output kW	12.49	12.01	9.45	4.98	
	C.O.P.	0.99	1.07	1.16	1.27	
III 35-60 °C	Engine Shaft Power kW	2.20	2.07	1.69	1.26	0.86
	Gross Gas Input Power kW	13.02	12.25	10.00	7.46	5.09
	Heat Pump Output kW	5.03	5.75	5.58	5.00	3.98
	Heat Reclaimed kW	8.07	7.60	6.20	4.63	3.16
	Total Heat Output kW	13.10	13.35	11.78	9.63	7.14
	C.O.P.	1.01	1.09	1.18	1.29	1.40
IV 45-65 °C	Engine Shaft Power kW	2.07	1.43	No results possible - engine stalled		
	Gross Gas Input Power kW	12.25	8.46			
	Heat Pump Output kW	4.40	3.69			
	Heat Reclaimed kW	7.35	5.08			
	Total Heat Output kW	11.75	8.77			
	C.O.P.	0.96	1.04			
V 50-70 °C	Engine Shaft Power kW	1.81	1.02	No results possible - engine stalled		
	Gross Gas Input Power kW	10.71	6.04			
	Heat Pump Output kW	3.64	2.50			
	Heat Reclaimed kW	6.21	3.50			
	Total Heat Output kW	9.85	6.00			
	C.O.P.	0.92	0.99			

*N.B. Groups I to V refer to different sets of inlet and outlet water temperatures.

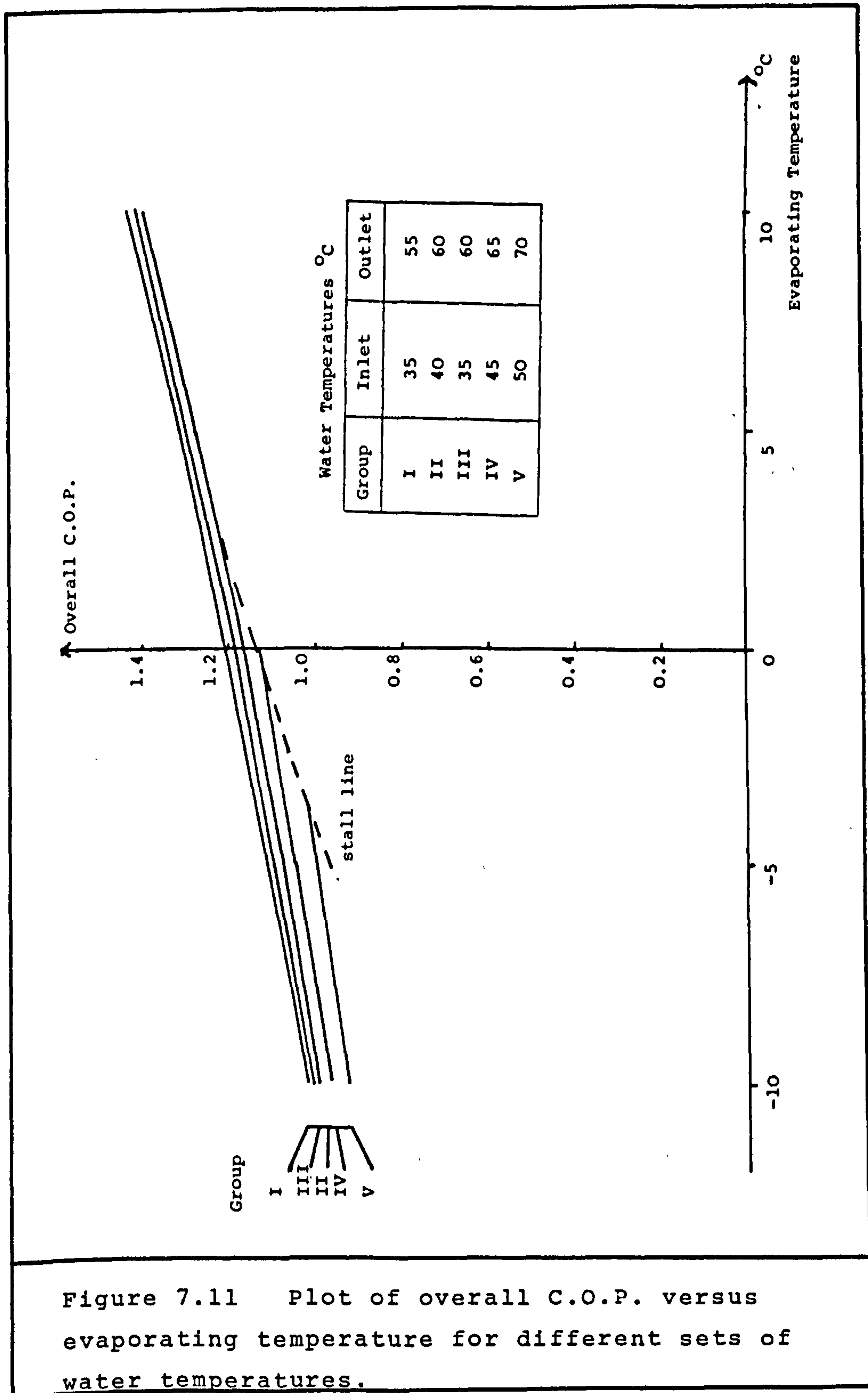


Figure 7.11 Plot of overall C.O.P. versus evaporating temperature for different sets of water temperatures.

7.9 Control strategy Tests

Modulation

Throttling proved impossible with our particular single cylinder engine. The engine could not run at anything less than full throttle due to problems of carburation, as explained earlier.

Intermittency Tests

We can only compare the results of the intermittency tests within a particular set, since the experiments were carried out on different days, under different operating conditions and no further time was available at the end of the project. The results are therefore inconclusive. It can be seen from Table 7.13 that if cycling for short time intervals (as in Set A), it is better to turn off the water pump, and retain as much heat as possible.

Table 7.13 Results of Intermittency Tests.

		Water Pump on, Heat Pump cycling	Heat Pump & Water Pump cycling	Continuous Running
Set A: 30 min runs, cycling every 5 mins.	C.O.P	0.94	1.16	1.17
Set B: 30 min runs, cycling every 10 mins.	C.O.P.	1.05	1.07	1.17
Set C: 45 min runs, cycling every 15 mins.	C.O.P.	1.08	1.07	1.10

Also, a short cycle time gives the closest result to that obtained by continuous running, which suggests a good strategy for operating this particular machine. It should be

noted that throughout the testing we believed that our heat pump reached its steady state after about 15 mins and all measurements were taken after allowing for this. However as Table 7.13 indicates, running continuously for 45 mins gave a lower C.O.P. which suggested we may have suffered from transients and a different control strategy may have been better. Unfortunately we could not test this hypothesis as our running time was limited by our supply of water at a fixed temperature and the whole project being curtailed prematurely. Figure 7.11 shows the stall line.

7.10 Noise Measurements

An important factor for consideration when examining the acceptability of a gas engine driven heat pump system is the noise of such a machine. Most noise emanates from the bottom half of the unit, which contains the engine, compressor and condenser. These are all fitted onto a tubular steel frame, which sits on anti-vibration mounts. The top half of the unit consisting of the evaporator, fan and electronics and is linked to the bottom half by flexible pipes. When the machine is in operation the bottom half of the unit sits inside an acoustic enclosure as shown in Plate I. In the original design, the centrifugal clutch was found to be very noisy. The replacement electromagnetic clutch remedied the problem however. Also, non-alignment of the fan shaft caused an excessive vibration. With these modifications carried out, plus strapping down various pipe runs, and the use of the acoustic enclosure, the noise level was reduced to 60 dBA at 1 metre and was not restricted to bass frequencies which compares favourably with a great many electric heat pumps.

CHAPTER 8 CONCLUSIONS

In general we can conclude that the gas engine driven heat pump worked almost as well as our design calculations had indicated and that the reasons for any drop in expected performance are well understood. For example, at an average ambient temperature of 6°C, our design calculations led us to expect an output of 17 kW with an overall C.O.P. of 1.25, whereas in practice we achieved (from the mean of the results) an output of 14 kW with an overall C.O.P. of 1.10 (Figure 7.9).

A comparison of the design heat pump C.O.P.'s and those achieved (Figure 7.7) show that the measured results are only slightly less than those expected, which confirms our overall design of the refrigeration circuit. In addition we found that after modification the starting and stopping procedures worked satisfactorily and that the safety precautions were adequate.

The main cause of the drop in performance experienced can be accounted for by the inefficiency of the engine. Earlier engine tests with British Gas at Isleworth indicated that an efficiency of 20% was possible, whereas our maximum efficiency was only 16.9%. This was due to inadequate gas regulation - the air/fuel ratio mixture was not steady, but fluctuated due to the combination of our particular single cylinder engine and the gas regulator. It should be noted that this problem is engine specific and should not be taken

as a discouragement of any future work.

The compressor was found to work very reliably and we experienced no problems with this part of the system. (Electric heat pumps are very prone to problems in this area) 80% of the manufacturer's claimed performance was achieved (Figure 7.5) and most of the 20% loss can be explained by a pressure drop in the system due to narrow piping with many bends. This could therefore be easily remedied by using larger diameter piping.

Results from the evaporator unit indicate that only half the evaporator face area was used. This was due to poor airflow characteristics (caused by the recirculation ducting fitted to enable us to achieve low evaporating temperatures in the laboratory). However, this effect caused no lack of performance since the evaporator had been deliberately oversized in our design calculation. It should also be noted that the fan drive from the engine worked well, and therefore on energy grounds, this would seem a better alternative to using an electric motor.

As far as heat reclamation was concerned the measured results were in good agreement, in percentage terms, with those predicted (Table 7.8). The only notable difference was in increased radiation loss, due to a poor performance by the exhaust condenser. However, the main exhaust gas heat exchanger worked better than expected. This is a simple concentric 'tube in tube' construction, designed to normal

heat transfer calculations.

Using the hot gas bypass for defrost worked quickly and efficiently and seems as good a way as any of achieving defrost, however there was no time available in the project to determine its effect on the overall C.O.P.

The control strategy tests proved, unfortunately, to be inconclusive. Modulation by throttling, was impossible since no running could be achieved at anything less than full throttle due to the engine specific problems mentioned earlier. The intermittency tests indicated that when cycling for a short time, it was better to turn off the recirculating water and retain as much of the heat in the engine as possible, whereas if a longer cycle time is employed some useful heat can be picked up by leaving the water pump on after the engine has been turned off.

It should be pointed out that the particular control strategy chosen, with the engine continuously running on full throttle (to maximise efficiency) and running speed is determined by load, worked very well. Over the normal operating range the compressor load slows down the engine until a point of equilibrium is reached, denoted by the cutting of the power curve by the appropriate extraction line (Table 7.5 and Figure 7.4). At very low evaporating conditions the engine governor prevents an increase in speed by closing the throttle; and at high evaporating conditions the compressor suction pressure regulator maintains a constant power demand.

Experiments with different sets of running (water) temperatures showed the expected behaviour (Figure 7.11): as the engine head temperature increases, so the performance decreases. However, much higher water temperatures are attainable with such a system as this, than with electric heat pumps. It should be noted that the stall line seen in the figure is again engine specific.

In designing a larger unit we would now have full confidence in using a reputable manufacturer's data for design purposes, taking account of the operating conditions and limitations. Also, using a larger engine such as the Ford 2274E, a better efficiency would be expected. Heat reclamation could also be improved by utilising the heat from inside any acoustic enclosure used.

The economic assessment carried out would indicate that the gas engine driven heat pump is a fuel saver wherever gas (or indeed liquid fuel) is burnt to produce low temperature heat, and that this fuel saving results in a lower cost of useful heat to the consumer. Using a typical running time of 2000 hours per year, and assuming no increase in fuel prices, the simple payback time for a 100 kW heat pump, against a 100 kW gas boiler is 7.44 years (Figure 4.1). Such a machine could be used to heat a group of houses or any commercial industrial heat or hot water demand of that size for which there would appear to be a large potential market.

Further, from Net Present Cost (NPC) calculations (Figures

4.2 and 4.3), as the discount rate decreases below 20% (the internal rate of return, IRR), the economic case for a large heat pump system becomes greater. Obviously an increase of fuel prices will lead to an even greater IRR. Figure 4.3 shows that for a modest fuel price increase of 4% per annum the internal rate of return is 25%.

As explained in the introduction on page 12 many of the questions raised during the monitoring phase of the project had to be left unanswered, although the basic objectives of the project were achieved. The major technical queries remaining are of durability and performance over long time periods and logically the next stage of research would have been to construct a larger unit to heat a small group of houses (or any other suitable heat demand) for at least one complete heating season.

Postscript

It is interesting to note that although this work was completed in 1980, there are still (end 1983) only 20 gas engine driven heat pumps in operation in the U.K., mostly in industry and buildings, with the highest proportion in swimming pools. As stated earlier, the work described in this thesis was one of the first to be funded by the Department of Energy under its R.D and D programme which is currently supporting 9 of the above 20 systems.

PHOTOGRAPHIC PLATES

Table 8.1 Key to Photographic Plates.

In the ten black and white plates which follow, the numbers refer to the components listed below :

[illegible]

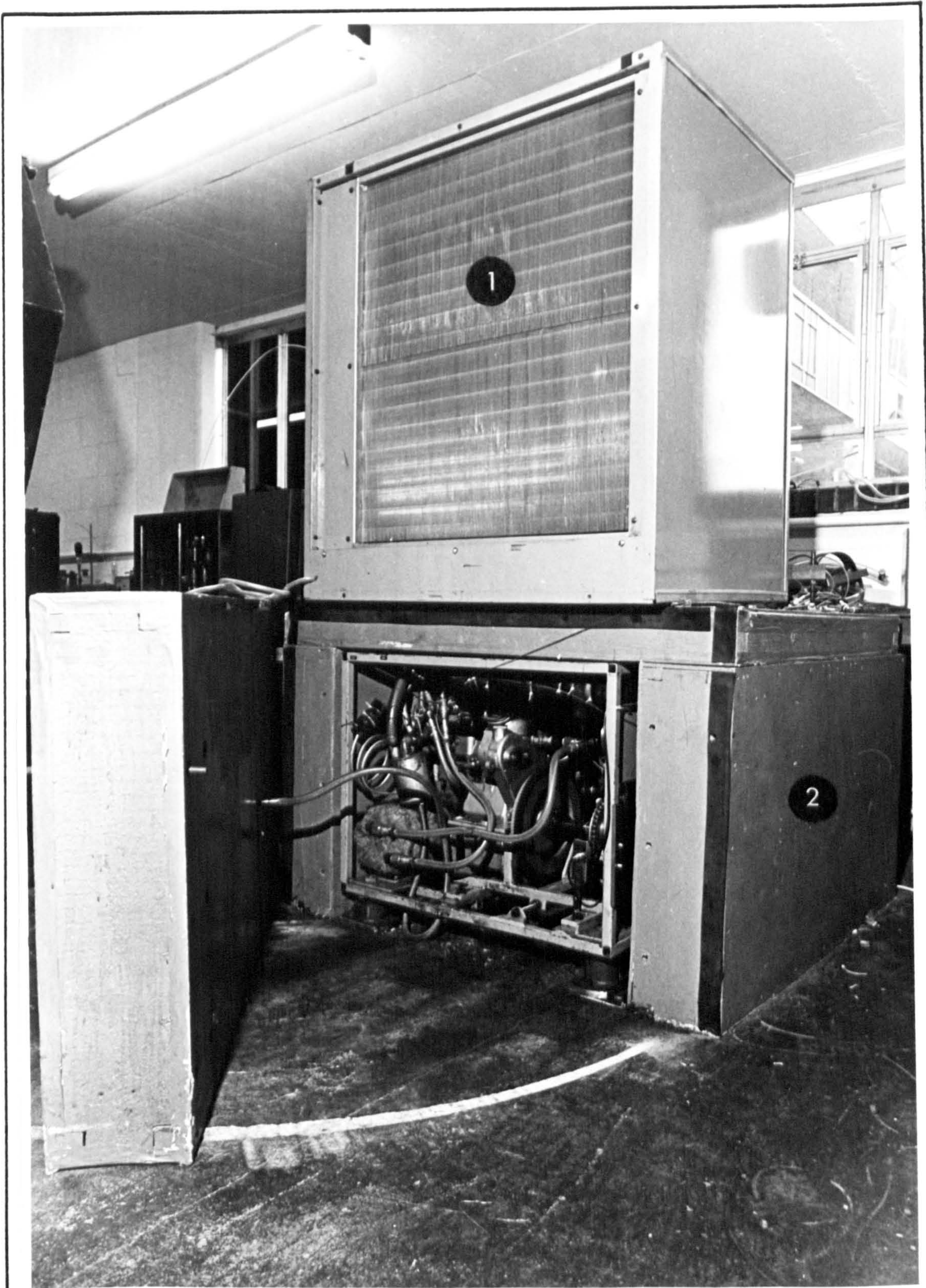


Plate I : General view of gas engine driven heat pump inside its acoustic enclosure.

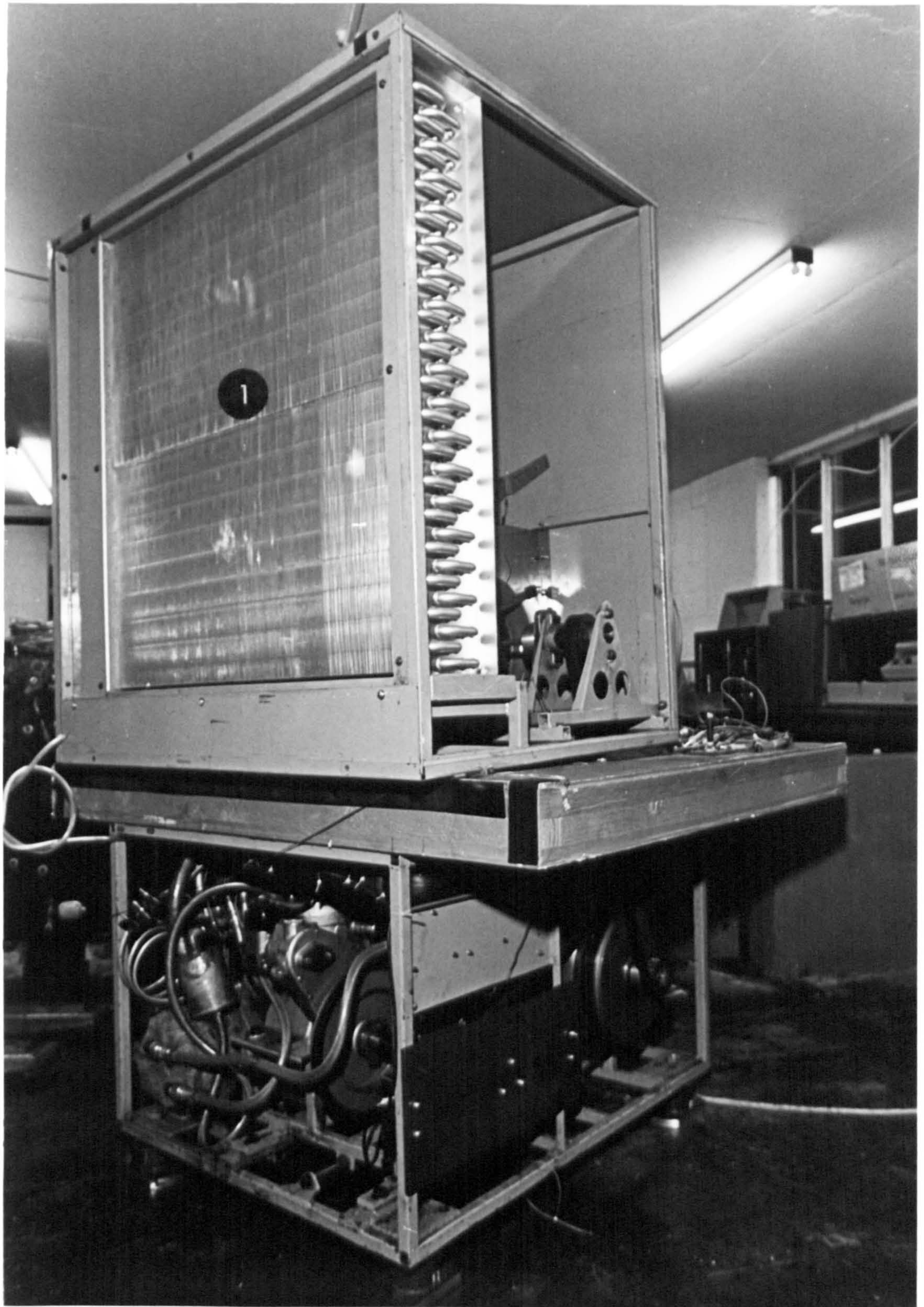


Plate II : General view of heat pump unit outside of its enclosure.

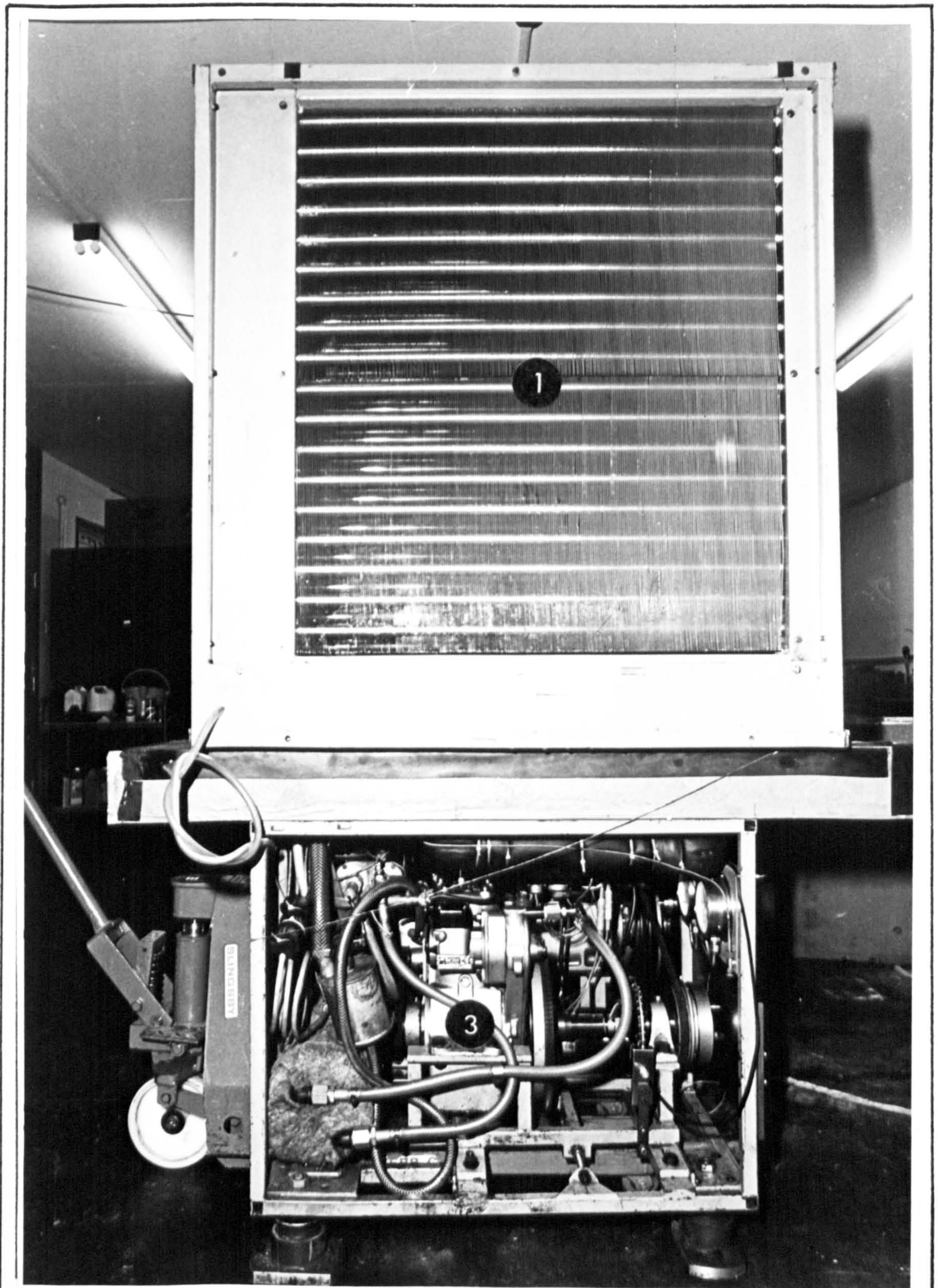


Plate III : Front view of heat pump unit outside its enclosure.
Evaporator unit above and engine assembly below.

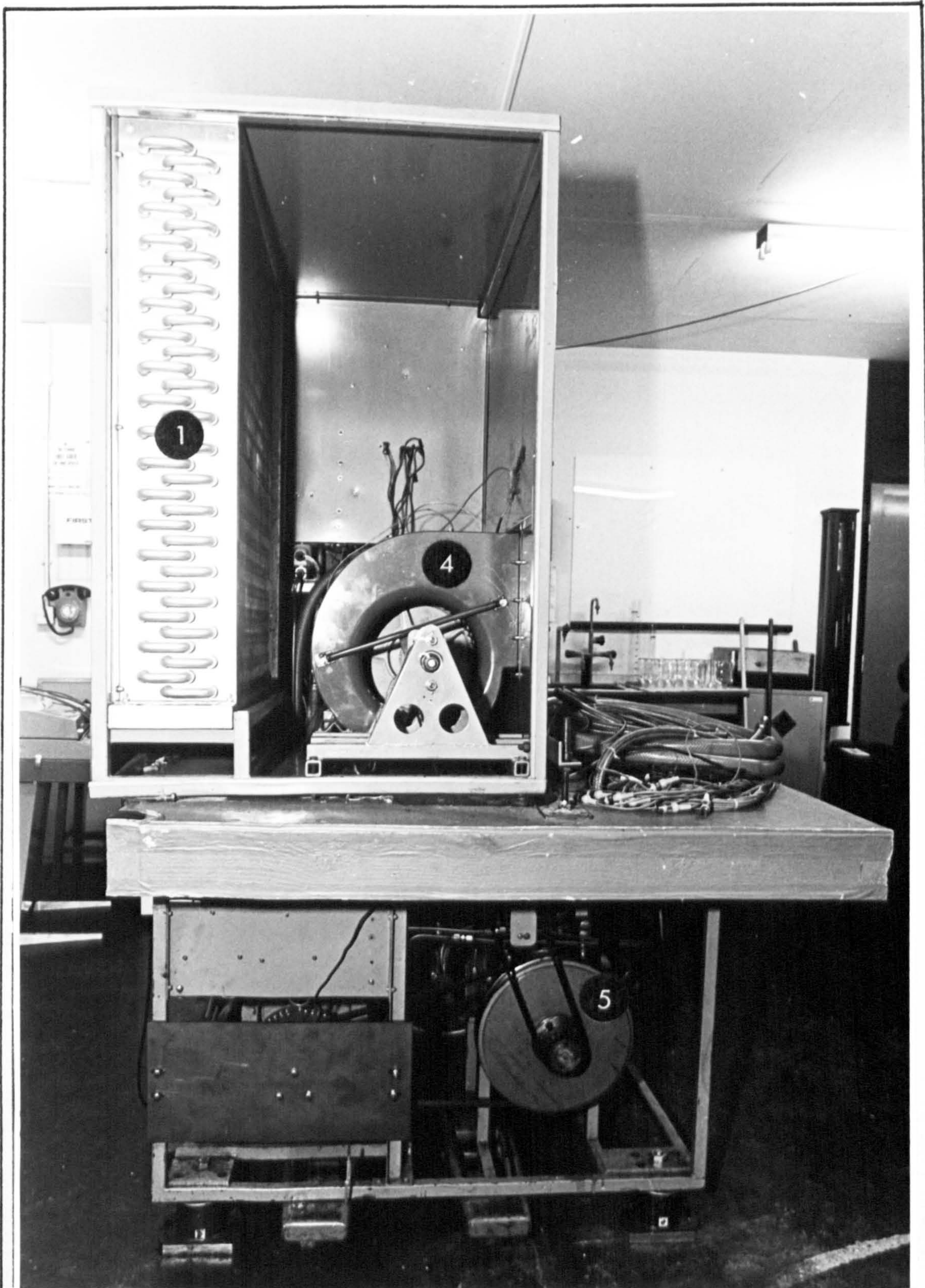


Plate IV : Side view of the heat pump unit with side panel removed showing centrifugal fan, belt driven from the compressor.

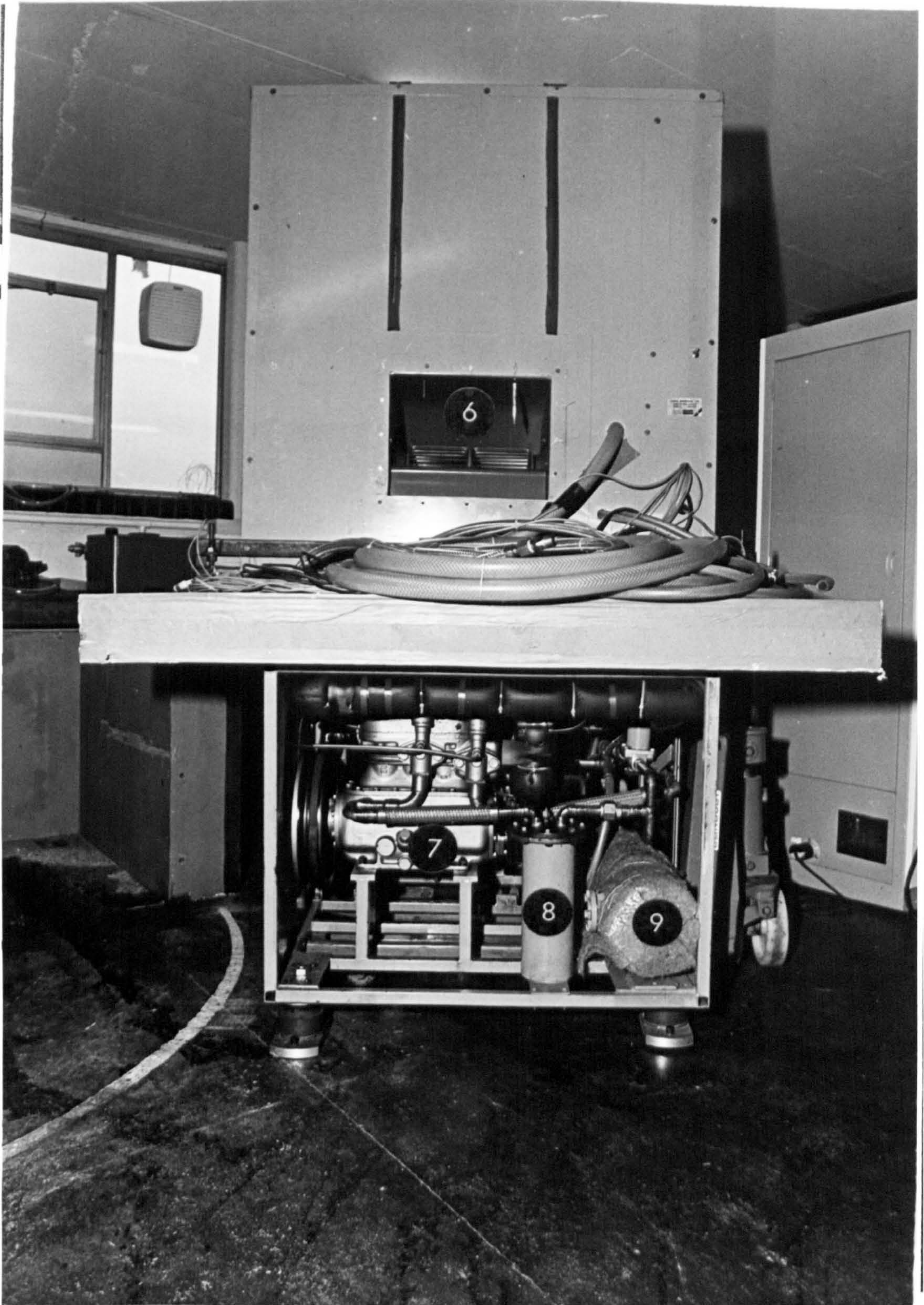


Plate V : Rear view of the heat pump unit, outside of the acoustic enclosure, showing the centrifugal fan (above), and the compressor and oil separator (below).

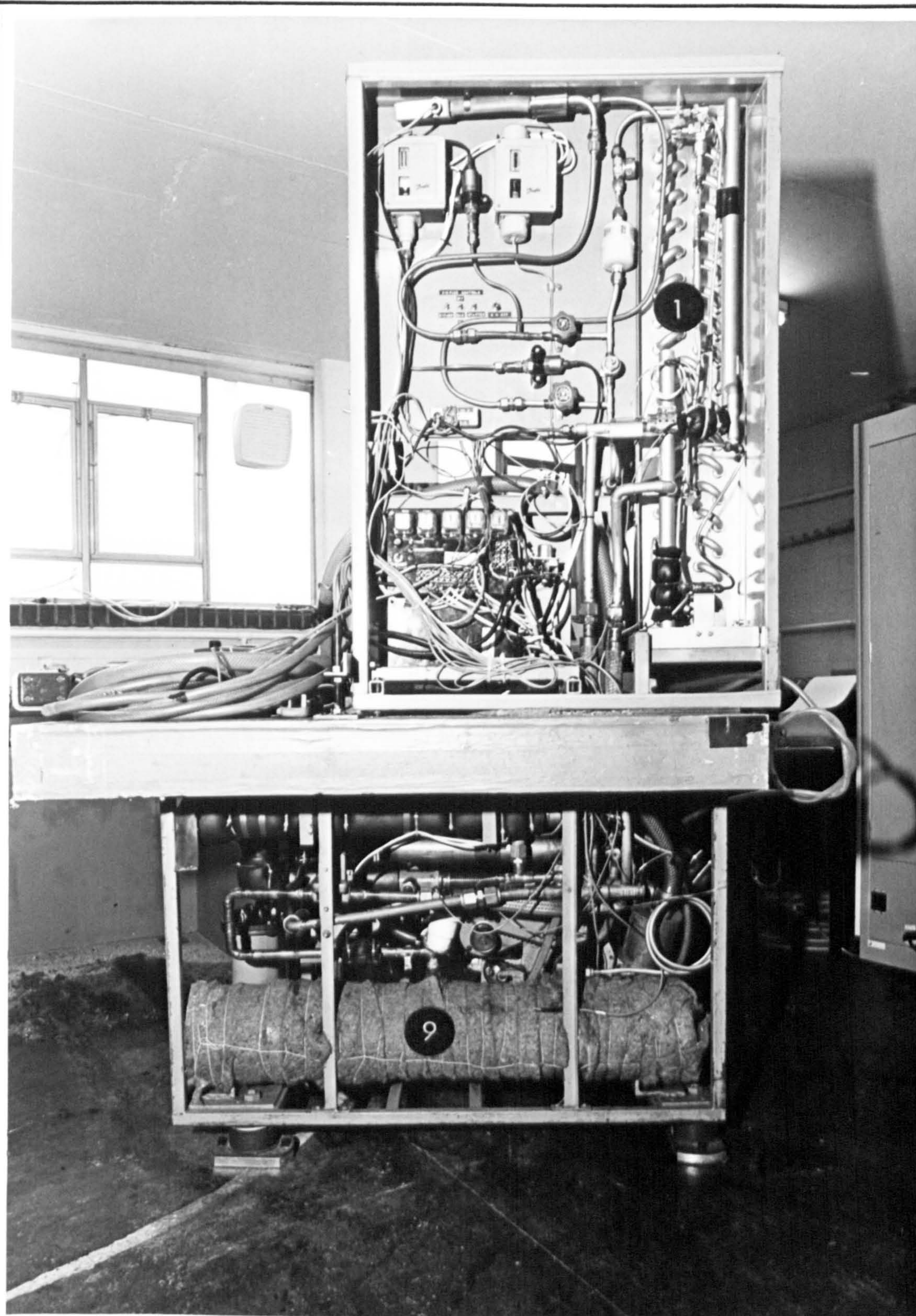


Plate VI : Side view of the heat pump unit, with side panel removed to show the control circuitry and refrigeration pipework (above), and condenser (below).

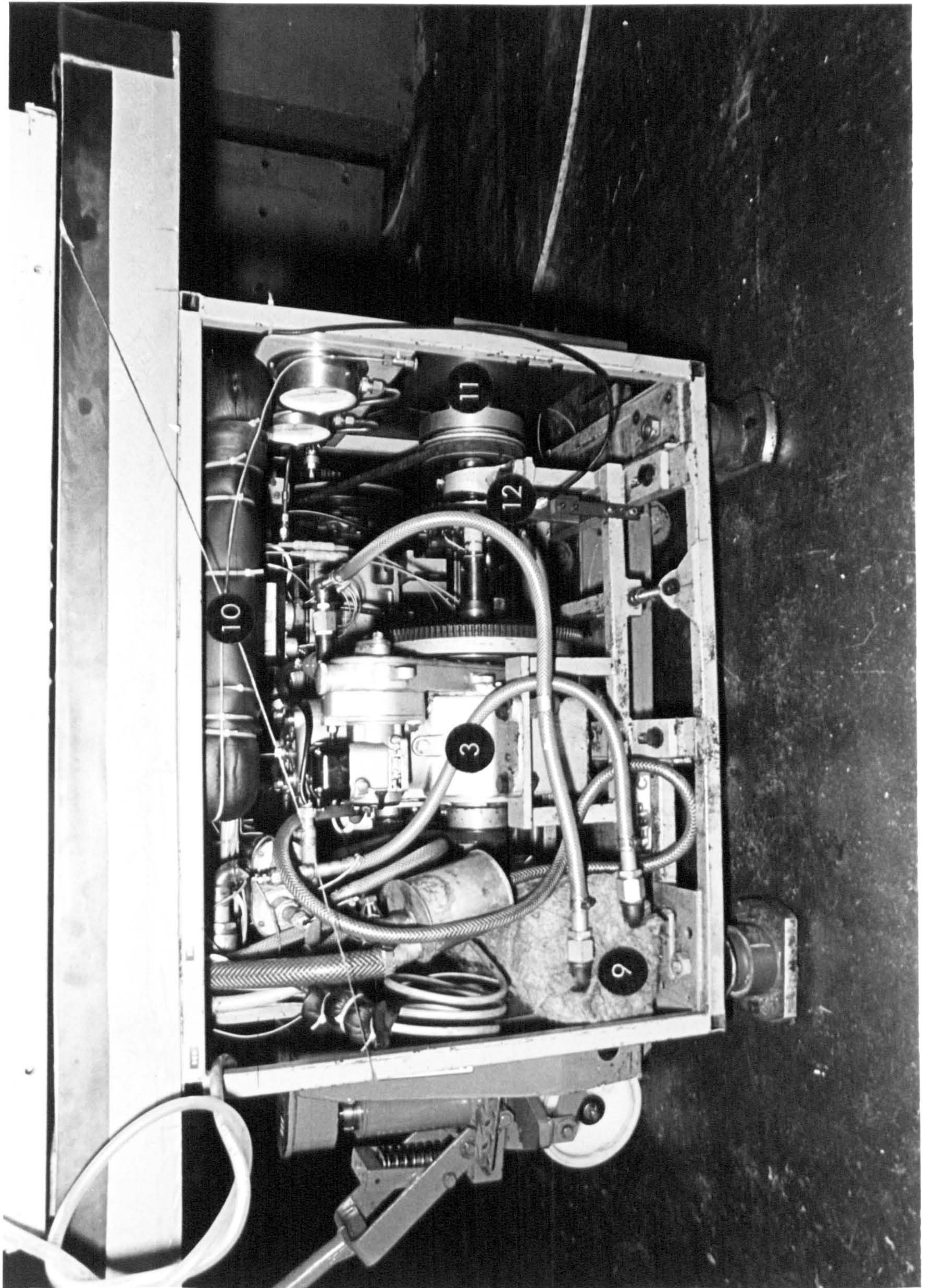


Plate VII : View of bottom half of unit showing engine assembly.

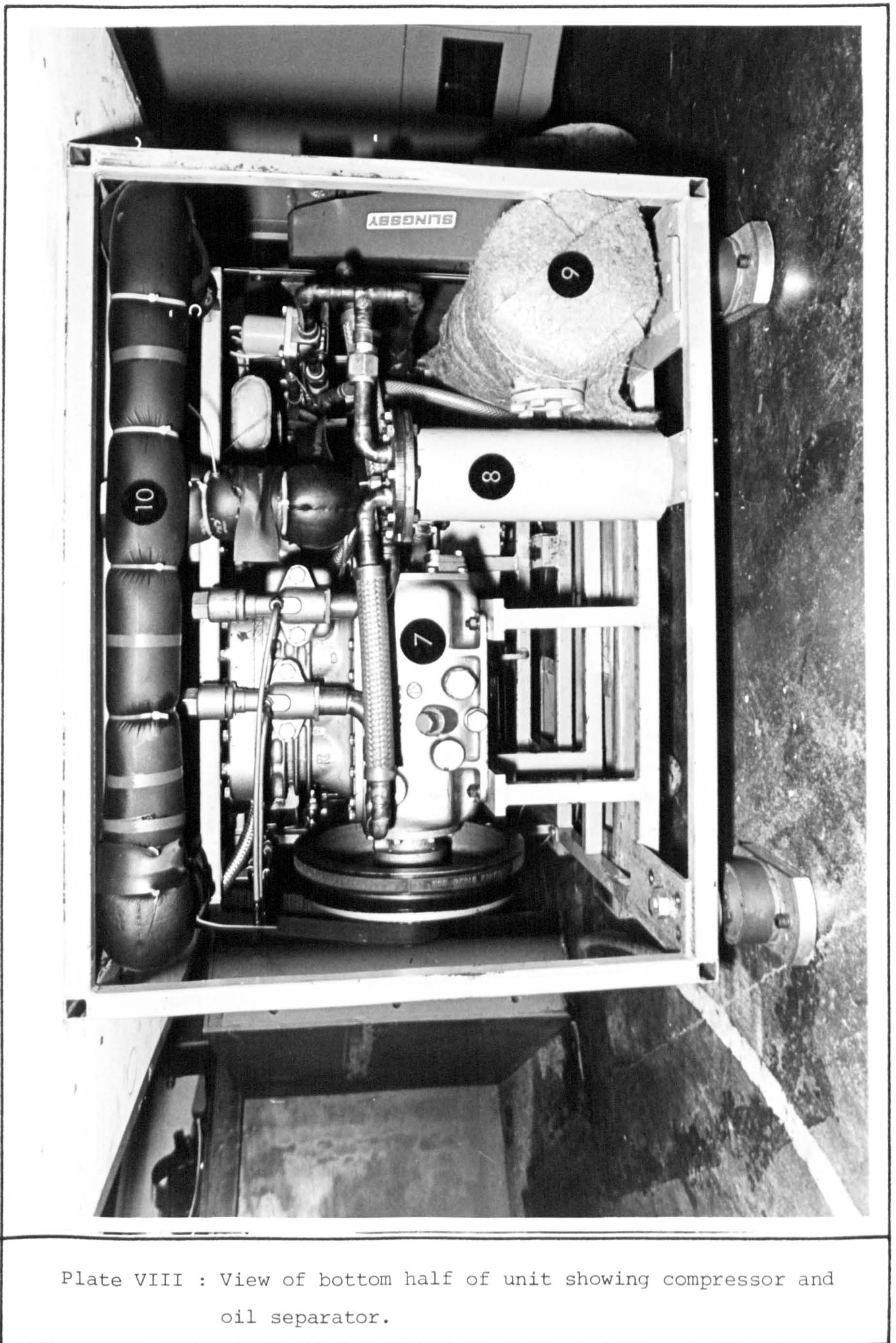


Plate VIII : View of bottom half of unit showing compressor and oil separator.

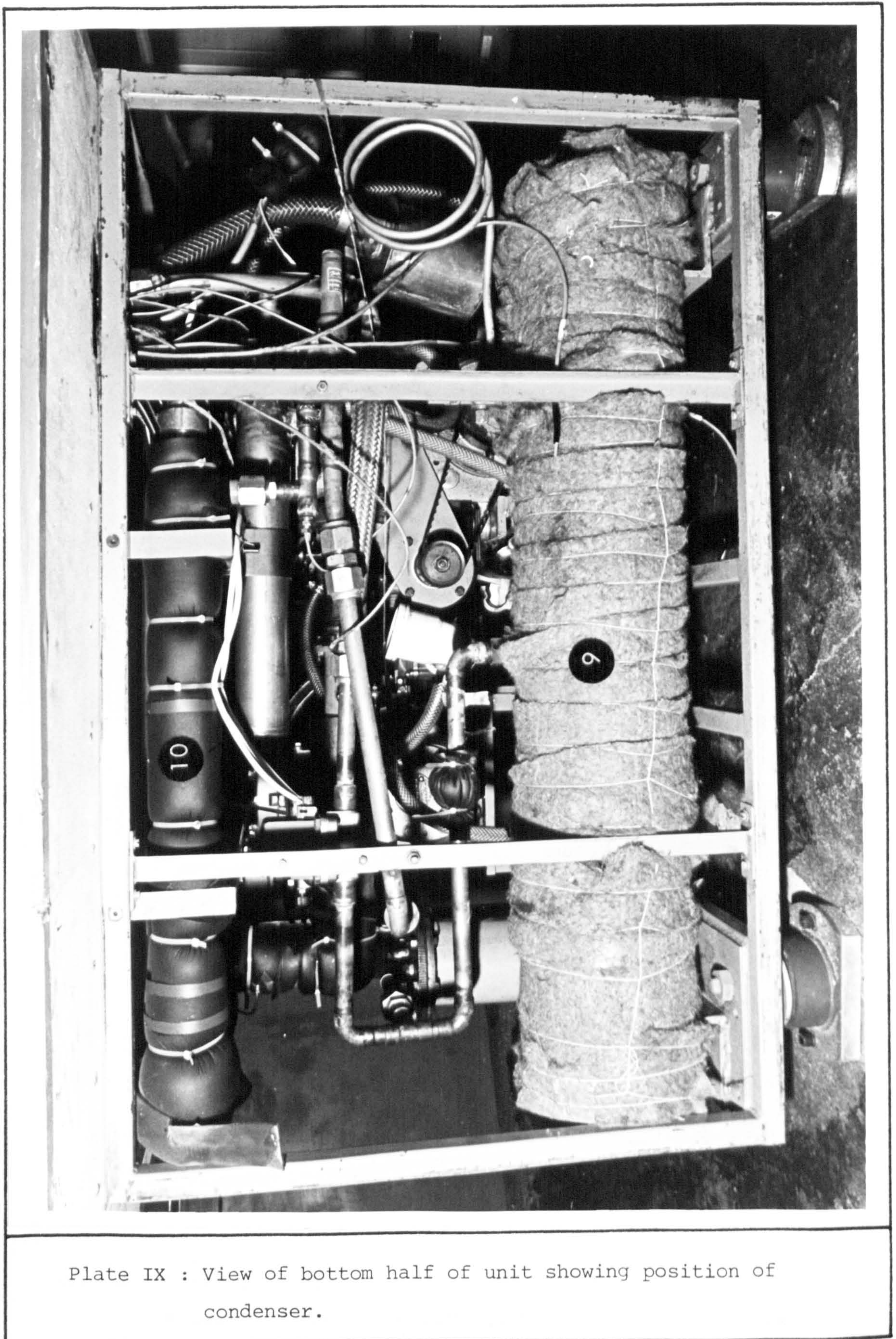


Plate IX : View of bottom half of unit showing position of condenser.

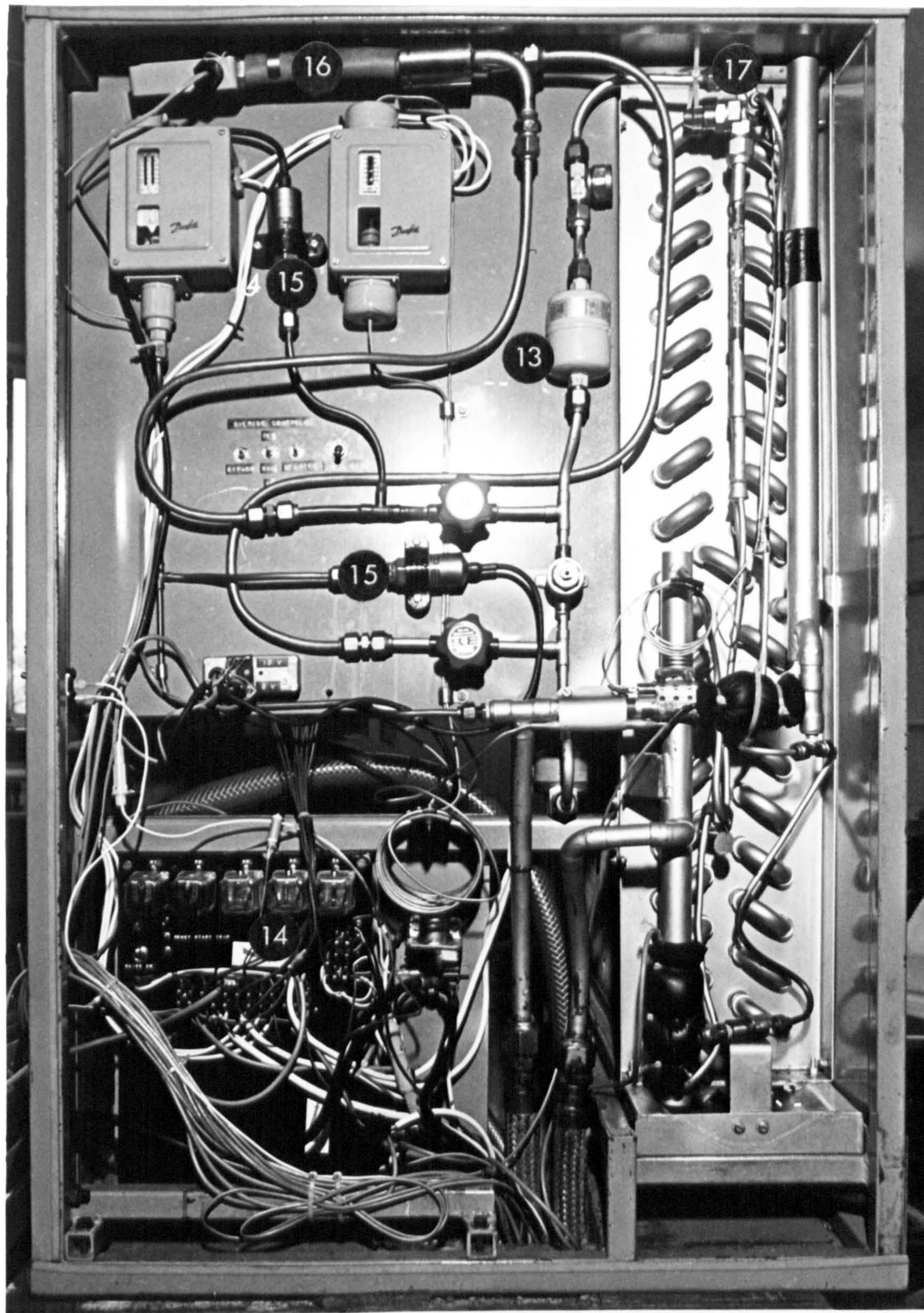


Plate X : View of upper half of heat pump unit (side panel removed) showing control circuitry and refrigeration pipework.

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CHAPTERS 6 AND 7

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Appendix I

Computer program to calculate the heat loss of an MKDC house.

```

INPUT NUMBER OF CARDS TO BE READJOB CM14 2502 HOUSE
ROUTE OPENUNIV
PHOENIX
FTGICLG (PROGRAM=XH) FT05F001=XH#) LISTC=SOURCE
COMMON/DD/NOROOM,TEMPI(20),OUTPUT(20),THEAT(20),TUAREA(20),TEMOUT,
CTEMGRD,RATE,D,SP,RANGE,AREA(20,15),UVAL(20,15),ITYPE(20,15),HT(20,
C15),UAREA(20,15),VENT(20),OUTOLD(20),NOUNIT(20),RLOSS(20)
CALL READ
CALL CALC
500 CALL LOSS
DO 60 I=1,NOROOM
TEMPI(I)=(OUTPUT(I)-THEAT(I))/TUAREA(I)
WRITE(6)TEMPI,OUTPUT
OUTOLD(I)=OUTPUT(I)
60 CONTINUE
CALL LOSS
DO 70 I=1,NOROOM
IF(ABS(OUTPUT(I)-OUTOLD(I)).LE.RANGE)GO TO 500
70 CONTINUE
CALL WRITE
STOP
END
SUBROUTINE READ
COMMON/DD/NOROOM,TEMPI(20),OUTPUT(20),THEAT(20),TUAREA(20),TEMOUT,
CTEMGRD,RATE,D,SP,RANGE,AREA(20,15),UVAL(20,15),ITYPE(20,15),HT(20,
C15),UAREA(20,15),VENT(20),OUTOLD(20),NOUNIT(20),RLOSS(20)
C READ DATA FOR CALCULATION OF HEAT LOSS
C READ IN OUTSIDE TEMP.
READ (5,50) TEMOUT
50 FORMAT(F10.0)
WRITE (6,40) TEMOUT
40 FORMAT (' OUTSIDE TEMP.=',F4.2,'C')
TEMGRD=TEMOUT
TOTAL=0.0
C READ IN NO. OF ROOMS
READ (5,100) NOROOM
100 FORMAT(I10)
WRITE (6,30) NOROOM
30 FORMAT(' NO.OF ROOMS=',I2)
READ(5,5)RATE,D,SP
5 FORMAT(3F10.0)
WRITE(6,12)RATE,D,SP
12 FORMAT(' RATE=',F4.2,'PER HR',3X,'D=',F6.3,3X,'SP=',F6.2)
READ(5,80)RANGE
80 FORMAT(F10.0)
DO 10 I= 1,NOROOM
READ (5,110) NOUNIT(I),TEMPI(I)
110 FORMAT(I10,F10.0)
IF(TEMPI(I).NE.0.0)GO TO 6
TEMPI(I)=TEMOUT
6 WRITE(6,20)NOUNIT(I),TEMPI(I)
20 FORMAT(' NO. OF UNITS=',I2,5X,'INTERNAL TEMP.=',F6.2)
VENT(I)=0.0
OUTPUT(I)=0.0
RLOSS(I)=0.0
TUAREA(I)=0.0
THEAT(I)=0.0
NUM= NOUNIT(I)
WRITE(6,15)
15 FORMAT(15,'AREA',I15,'U VALUE',I25,'TYPE OF CONSTRUCTION',I50,'HEI
CGHT')
DO 10 J=1,NUM
READ(5,120)AREA(I,J),UVAL(I,J),ITYPE(I,J),HT(I,J)
120 FORMAT(2F10.0,I10,F10.0)
WRITE(6,25)AREA(I,J),UVAL(I,J),ITYPE(I,J),HT(I,J)
25 FORMAT(I3,F8.4,I15,F6.3,I30,I2,I50,F6.3)
10 CONTINUE
RETURN
END
SUBROUTINE CALC
COMMON/DD/NOROOM,TEMPI(20),OUTPUT(20),THEAT(20),TUAREA(20),TEMOUT,
CTEMGRD,RATE,D,SP,RANGE,AREA(20,15),UVAL(20,15),ITYPE(20,15),HT(20,
C15),UAREA(20,15),VENT(20),OUTOLD(20),NOUNIT(20),RLOSS(20)
DO 11 I=1,NOROOM
NUM=NOUNIT(I)
DO 11 J=1,NUM
UAREA(I,J)=AREA(I,J)*UVAL(I,J)
TUAREA(I)=TUAREA(I)+UAREA(I,J)
11 CONTINUE
RETURN
END

```

Cont'd


```

SUBROUTINE LOSS
COMMON/DO/NOROOM,TEMPI(20),OUTPUT(20),THEAT(20),TUAREA(20),TEMOUT,
CTEMGRD,RATE,D,SP,RANGE,AREA(20,15),UVAL(20,15),ITYPE(20,15),HT(20,
C15),UAREA(20,15),VENT(20),OUTOLD(20),NOUNIT(20),RLOSS(20)
C CALCULATE HEAT LOSS(GAIN) ROOM BY ROOM
DO 21 I=1,NOROOM
NUM=NOUNIT(I)
DO 21 J=1,NUM
IT=ITYPE(I,J)
QHEAT=UAREA(I,J)*TEMPI(I)
THEAT(I)=THEAT(I)+HEAT(IT,I,J)
OUTPUT(I)=OUTPUT(I)+QHEAT-THEAT(IT,I,J)
VENT(I)=VENT(I)+(RATE/3600.0)*D*SP*AREA(I,J)*HT(I,J)*(TEMPI(I)-
CTEMOUT)
21 CONTINUE
RETURN
END
FUNCTION HEAT(IT,I,J)
COMMON/DO/NOROOM,TEMPI(20),OUTPUT(20),THEAT(20),TUAREA(20),TEMOUT,
CTEMGRD,RATE,D,SP,RANGE,AREA(20,15),UVAL(20,15),ITYPE(20,15),HT(20,
C15),UAREA(20,15),VENT(20),OUTOLD(20),NOUNIT(20),RLOSS(20)
GO TO (1000,2000,3000),IT
WRITE(6,130) I,J
130 FORMAT (' COCK UP IN TYPE(I=',I2,',J=',I2,',)')
1000 HEAT=UAREA(I,J)*TEMOUT
RETURN
2000 HEAT=UAREA(I,J)*TEMGRD
RETURN
3000 DO 41 II=1,NOROOM
C MAX. NO. OF UNITS
NOMAX=NOUNIT(II)
DO 41 JJ=1,NOMAX
IF(AREA(I,J).EQ.AREA(II,JJ).AND.I.NE.II)GO TO 3050
41 CONTINUE
WRITE(6,140) I,J
140 FORMAT (' NO MATCH AREA,(I=',I2,',J=',I2,',)')
GO TO 1000
3050 HEAT=UAREA(I,J)*TEMPI(II)
RETURN
END
SUBROUTINE WRITE
COMMON/DO/NOROOM,TEMPI(20),OUTPUT(20),THEAT(20),TUAREA(20),TEMOUT,
CTEMGRD,RATE,D,SP,RANGE,AREA(20,15),UVAL(20,15),ITYPE(20,15),HT(20,
C15),UAREA(20,15),VENT(20),OUTOLD(20),NOUNIT(20),RLOSS(20)
C WRITE RESULTS OF CALC. OF HEAT LOSS
TOTAL=0.0
WRITE(6,150)
150 FORMAT(1H1,'FABRIC LOSS',5X,'VENTILATION LOSS',5X,'ROOM HEAT LOSS',
C,5X,'ROOM TEMP.')
DO 31 I=1,NOROOM
RLOSS(I)=OUTPUT(I)+VENT(I)
WRITE(6,160)OUTPUT(I),VENT(I),RLOSS(I),TEMPI(I)
160 FORMAT(F10.4,T25,F10.4,T40,F10.4,T60,F10.4)
TOTAL=TOTAL+RLOSS(I)
31 CONTINUE
WRITE(6,170)
170 FORMAT(1H0,'***TOTAL HOUSE HEAT LOSS(GAIN)***')
WRITE(6,180) TOTAL
180 FORMAT(T8,F15.4,'WATTS')
RETURN
END

```

The program then prints out the rate of heat loss of the house as shown overleaf :

OUTSIDE TEMP.= .00C
 NO. OF ROOMS=15
 RATE=1.00PER HR U= 1.293 SP=993.00

FABRIC LOSS	VENTILATION LOSS	ROOM HEAT LOSS	ROOM TEMP.
460.4472	110.9332	571.3804	18.0000
863.0474	215.3953	1078.4427	18.0000
79.1010	37.7481	116.8491	18.0000
267.8981	72.8924	340.7905	18.0000
114.4824	34.5125	148.9950	18.0000
74.1640	37.7789	111.9630	18.0000
381.6291	41.7540	423.3831	18.0000
316.7554	110.9332	427.6886	18.0000
448.2670	215.3953	663.6622	18.0000
1429.2677	312.7700	1742.0377	20.0000
111.5856	72.8924	184.4780	18.0000
108.8957	41.7540	150.6497	18.0000
43.0272	34.5125	77.5397	18.0000
43.0272	37.7481	80.7753	18.0000
0.0000	37.7789	37.7789	18.0000

TOTAL HOUSE HEAT LOSS(GAIN)
 6156.4139WATTS

Appendix II

Computer program to show the estimated number of hours running time for a heat pump operating at various ambient temperatures over a 27 year period.

Using real weather data from a Meteorological Office tape for Heathrow the computer program determined a heating requirement for each day (0700-2200 hrs) of an 8 month heating season (Oct-May). The program then worked out how long the heat pump would have to operate to satisfy this demand and tested for the range of ambient temperatures it would be running under.

```

1  JJB CM14 3822,SAVE
2  ROUTE OPENUNIT
3  PJST CM14
4  TAPE9 CM1402
5  COMP 60 SECS
6  PIX
7  //MET DD UNIT=TAPE9,VOL=SER=CM1402,LABEL=(1,SL,,IN),DISP=(OLD,KEEP),
8  // DSN=CM14.2,DCB=(RECFM=VS,LRECL=392,BLKSIZE=396,DEN=3)
9  FTGICLG(FTGIFOO1=MET PROGRAM=XH) )
10 COMMON/BLOCK1/PRELIM(2),DATA(8,24)
11 EQUIVALENCE(INT(1),PRELIM(1))
12 INTEGER *2 INT(194),PRELIM,DATA
13 INTEGER HRS,THRS
14 DIMENSION TA(24),HRS(24),THRS(24)
15 HLOSS=0.0
16 H=0.0
17 P=0.0
18 DO 10 I=1,12
19   THRS(I)=0
20   10 CONTINUE
21   WRITE(6,100)
22   100 FORMAT(' *****DATA SHOWING NUMBER OF HOURS RUNNING AT VARIOUS
23   ' OUTSIDE TEMPERATURES*****',//)
24   WRITE(6,200)
25   200 FORMAT(' NOTE-MONTHS OF JUNE,JULY,AUGUST,SEPTEMBER OMITTED,
26   ' HEATING 0700 TO 2200, PUMP STARTING AT 1300',//)
27   WRITE(6,300)
28   300 FORMAT(' YEAR',IX,'T(C)',IX,'<-4',T20,'-4.1-(-2)',T30,'-2.1-0',T40
29   ' ,0.1-2',T50,'2.1-4',T60,'4.1-6',T70,'6.1-8',T80,'8.1-10',T90,
30   ' ,10.1-12',T100,'12.1-14',T110,'14.1-16',T120,'>16',//)
31   666 DO 20 I=1,12
32     20 HRS(I)=0
33   888 REA(I,END=999)INT
34     IYR=PRELIM(1)
35     IMTH=PRELIM(2)
36     ID=DATA(1,I)
37     IF(IMTH.EQ.6.OR.IMTH.EQ.7.OR.IMTH.
38     EQ.8.OR.IMTH.EQ.9)GO TO 888
39     DO 30 I=1,24
40       IF(DATA(2,I).LT.7.OR.DATA(2,I).GT.22)GO TO 30
41       TA(I)=DATA(5,I)/10.0
42       HLOSS=0.2801*(20.0-TA(I))
43       H=H+HLOSS
44     30 CONTINUE
45     L=0
46     P=H/15.0
47     L=IFIX(P)
48     IF(IFIX(P+0.5).GT.IFIX(P))L=L+1
49     I=12
50     DO 40 J=1,L
51       I=I+1
52       IF(TA(I).LT.-4.0)HRS(I)=HRS(I)+1
53       IF(TA(I).GE.-4.1.AND.TA(I).LE.-2.0)HRS(2)=HRS(2)+1
54       IF(TA(I).GE.-2.1.AND.TA(I).LE.0.0)HRS(3)=HRS(3)+1
55       IF(TA(I).GE.0.1.AND.TA(I).LE.2.0)HRS(4)=HRS(4)+1
56       IF(TA(I).GE.2.1.AND.TA(I).LE.4.0)HRS(5)=HRS(5)+1
57       IF(TA(I).GE.4.1.AND.TA(I).LE.6.0)HRS(6)=HRS(6)+1
58       IF(TA(I).GE.6.1.AND.TA(I).LE.8.0)HRS(7)=HRS(7)+1
59       IF(TA(I).GE.8.1.AND.TA(I).LE.10.0)HRS(8)=HRS(8)+1
60       IF(TA(I).GE.10.1.AND.TA(I).LE.12.0)HRS(9)=HRS(9)+1
61       IF(TA(I).GE.12.1.AND.TA(I).LE.14.0)HRS(10)=HRS(10)+1
62       IF(TA(I).GE.14.1.AND.TA(I).LE.16.0)HRS(11)=HRS(11)+1
63       IF(TA(I).GT.16.0)HRS(12)=HRS(12)+1
64     40 CONTINUE

```



```

65 H=0.0
66 IF(19.EQ.31.AND.INT1.EQ.12)GO TO 777
67 GO TO 666
68 777 WRITE(6,400)IYR,(HRS(J),J=1,12)
69 400 FORMAT(1X,14,T9,12(15,5X),/)
70 ON 50 I=1,12
71 50 THRS(I)=THRS(I)+HRS(I)
72 GO TO 666

73 999 WRITE(6,500)
74 500 FORMAT(' ***TOTAL HOURS RUNNING OVER 27 YEARS***',/)
75 WRITE(6,300)
76 WRITE(6,600)(THRS(J),J=1,12)
77 600 FORMAT(T9,12(17,3X))
78 STOP
79 END

```

Results

*****DATA SHOWING NUMBER OF HOURS RUNNING AT VARIOUS OUTSIDE TEMPERATURES*****

NOTE-MONTHS OF JUNE,JULY,AUGUST,SEPTEMBER OMITTED,

HEATING 0700 TO 2200, PUMP STARTING AT 1300

YEAR T(C)	<-4	-4.1-(-2)	-2.1-0	0.1-2	2.1-4	4.1-6	6.1-8	8.1-10	10.1-12	12.1-14	14.1-16	>16
1949	0	0	4	17	66	68	128	224	85	85	42	49
1950	0	1	46	93	81	49	120	179	105	95	42	37
1951	0	1	15	18	58	125	165	179	116	82	27	33
1952	0	8	29	44	171	168	87	107	86	90	31	31
1953	0	0	3	68	77	89	100	152	129	99	52	38
1954	2	22	36	39	82	76	103	172	124	99	33	33
1955	1	2	30	108	120	97	120	144	90	101	45	24
1956	8	29	80	67	104	65	105	172	100	97	32	34
1957	0	1	11	18	37	98	135	164	99	124	50	42
1958	0	2	27	33	120	98	152	151	80	81	52	25
1959	0	1	22	34	112	81	80	162	107	90	44	46
1960	0	0	9	37	91	121	100	128	147	81	39	32
1961	0	6	20	40	70	87	124	109	88	111	64	48
1962	13	20	46	77	81	129	182	123	101	61	42	19
→1963	25	45	104	141	112	70	76	95	91	107	41	21
1964	0	0	13	63	156	178	81	96	121	73	33	34
1965	0	0	19	68	98	179	132	103	91	81	47	27
1966	1	9	25	58	53	123	133	132	146	55	44	28
1967	0	5	22	51	53	62	172	192	120	77	33	17
1968	0	9	31	72	115	133	130	125	86	61	39	34
1969	0	1	22	98	124	111	157	101	94	46	41	41
1970	0	1	25	78	106	104	135	159	76	68	34	37
1971	0	9	18	26	71	97	135	191	99	62	28	47
1972	1	5	15	10	76	95	137	172	101	90	53	33
1973	0	0	2	34	71	119	162	182	116	67	41	19
1974	0	0	0	12	16	65	144	249	179	48	35	24
1975	0	1	8	24	33	146	166	167	130	78	39	16

TOTAL HOURS RUNNING OVER 27 YEARS

YEAR T(C)	<-4	-4.1-(-2)	-2.1-0	0.1-2	2.1-4	4.1-6	6.1-8	8.1-10	10.1-12	12.1-14	14.1-16	>16
	51	178	682	1428	2354	2833	3461	4130	2907	2209	1103	869

Appendix III

Computer program and results showing average hourly temperatures for each month of the year, over a 25 year timespan.

The following program was written to determine the average temperatures for each hour of the day during the month of January, over a 25 year timespan and using real weather data from a Meteorological Office tape for Heathrow. Similar programs were written for each month of the year and the results are shown in graphical form.

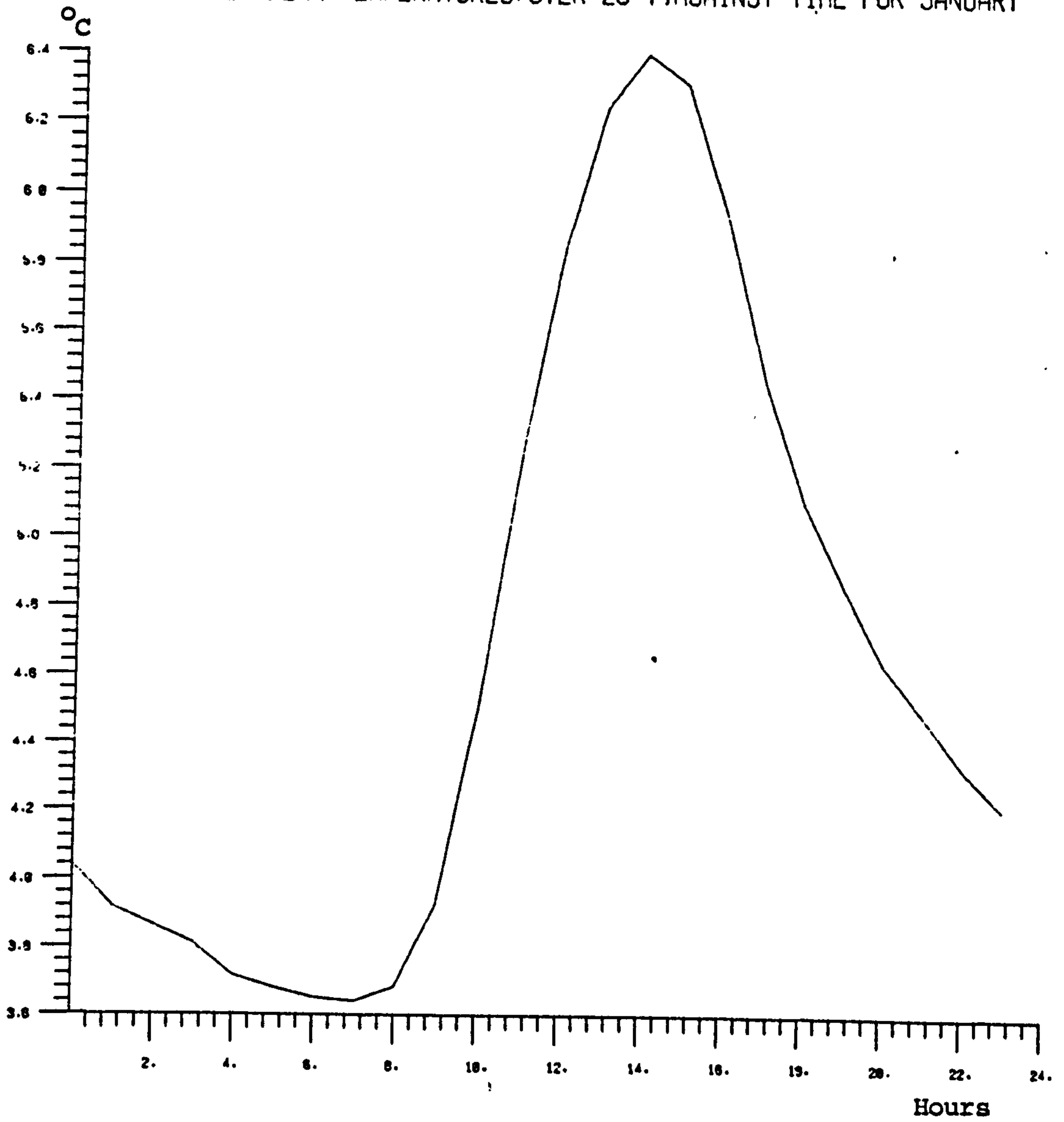
```

1  JOB CM14 3822 CM14JOB
2  ROUTE OPENUNIV
3  PLOTTER 100 RECORDS
4  P3ST MAIL
5  TAPE9 CM1402
6  COMP 30 SECS
7  PHX
8  //COMMOU4 DD SYSOUT=P
9  //MET DD UNIT=TAPE9,VOL=SER=CM1402,LABEL=(1,SL,,IN),DISP=(OLD,KEEP),
10 //   DSV=CM14.Z,DCB=(RECFM=VS,LRECL=392,BLKSIZE=396,DEN=3)
11 FTGICLG(FTOIFOO1=MET PROGRAM=%H1 FTO8FOO1=*4)LIBRARY=GRAPHICS.FORTLIB
12   REAL TIME(24),TMEAN(24),TOT(24)
13   INTEGER *2 INT(194),PRELIM,DATA
14   COMMON/BLOCK1/PRELIM(2),DATA(8,24)
15   EQUIVALENCE(PRELIM(1),INT(1))
16   DO 10 I=1,24
17     TOT(I)=0.0
18     TIME(I)=0.0
19   10 CONTINUE
20   J=1
21   D=31.0
22   888 REA(1,END=999)INT
23     IF(PRELIM(2).NE.J)GO TO 888
24     DO 20 I=1,24
25       TA=J.0
26       TA=DATA(5,I)/10.0
27       TOT(I)=TOT(I)+TA
28       TIME(I)=DATA(2,I)
29   20 CONTINUE
30   GO TO 888
31   999 WRITE(6,100)J
32   100 FORMAT(' MONTH ',I2)
33   WRITE(6,200)
34   200 FORMAT(T10,'TIME(HRS FROM MIDNIGHT)',T40,'MEAN TEMPERATURE
35     C(OVER 25 YRS)')
36   DO 30 I=1,24
37     TOT(I)=TOT(I)/D
38     TMEAN(I)=TOT(I)/25.0
39   WRITE(6,300)TIME(I),TMEAN(I)
40   300 FORMAT(T20,F6.2,T50,F8.2)
41   30 CONTINUE
42   CALL GFAR(TIME,TMEAN,24,0,0,'GRAPH OF MEAN TEMPERATURES(OVER 25 YRS
43     C)AGAINST TIME FOR JANUARY',61)
44   STOP
45   END

```

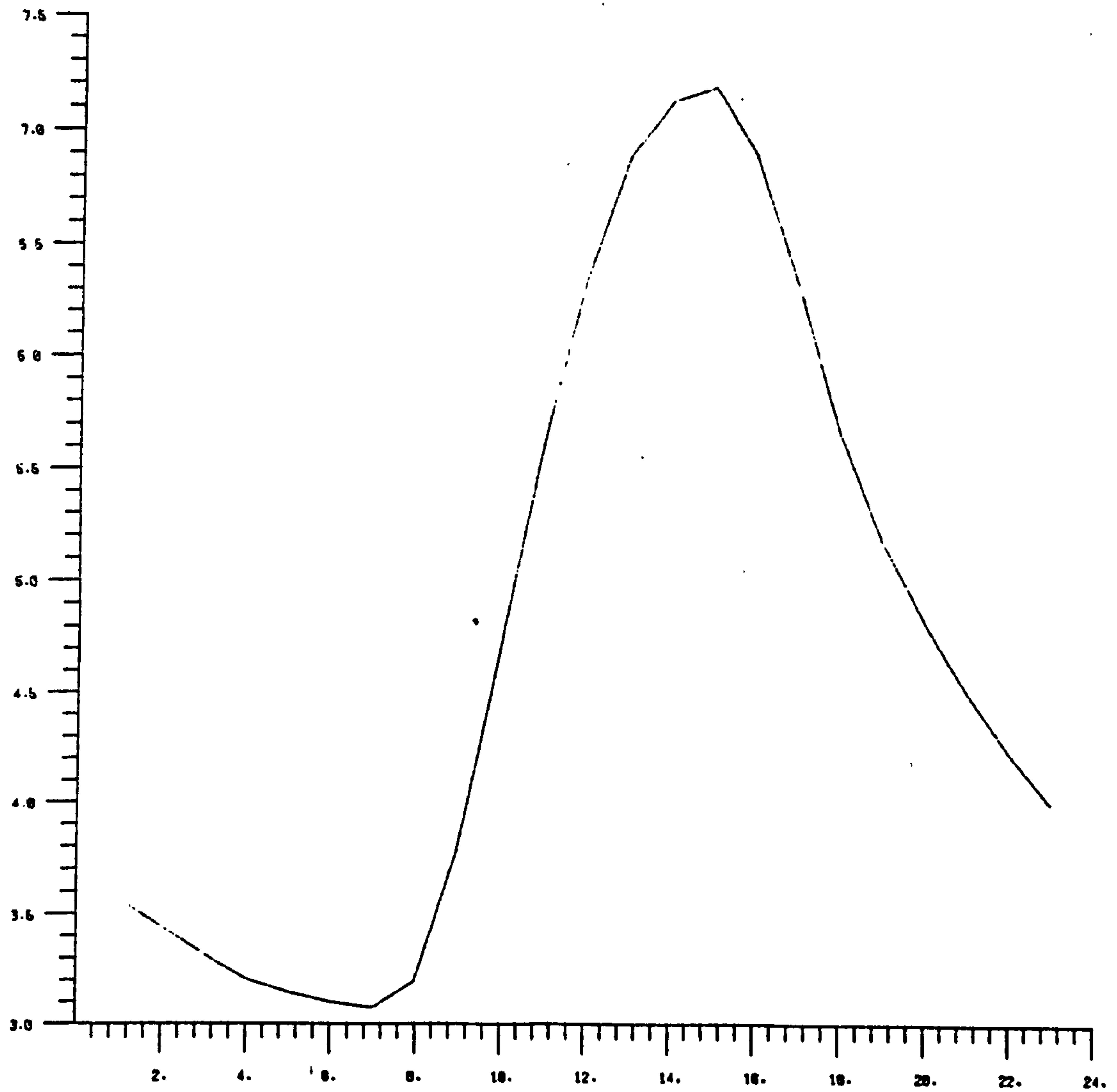
January

GRAPH OF MEAN TEMPERATURES (OVER 25 Y) AGAINST TIME FOR JANUARY



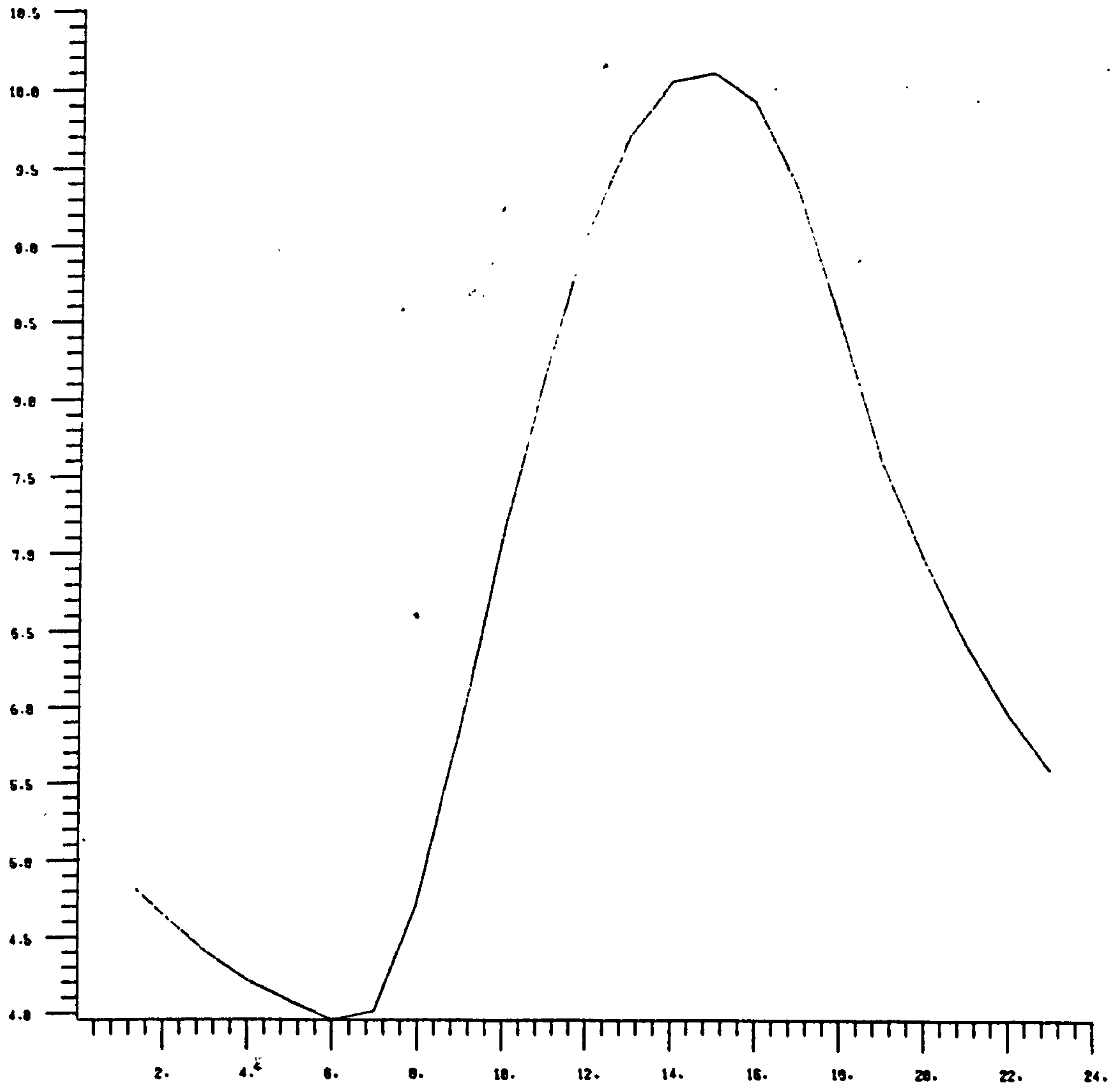
February

GRAPH OF MEAN TEMPERATURES(OVER 25 Y)AGAINST TIME FOR FEBRUARY



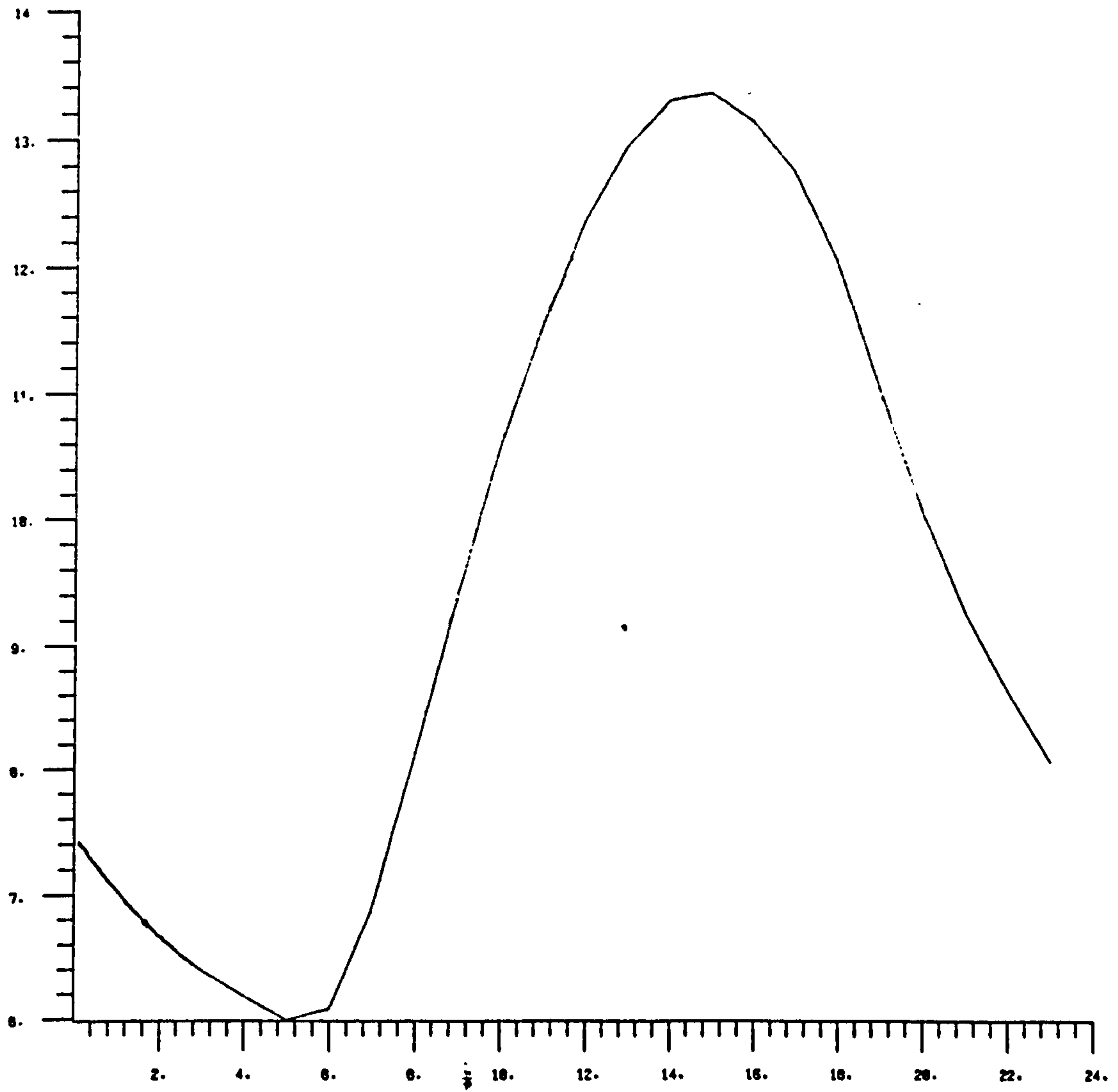
March

GRAPH OF MEAN TEMPERATURES (OVER 25 Y) AGAINST TIME FOR MARCH



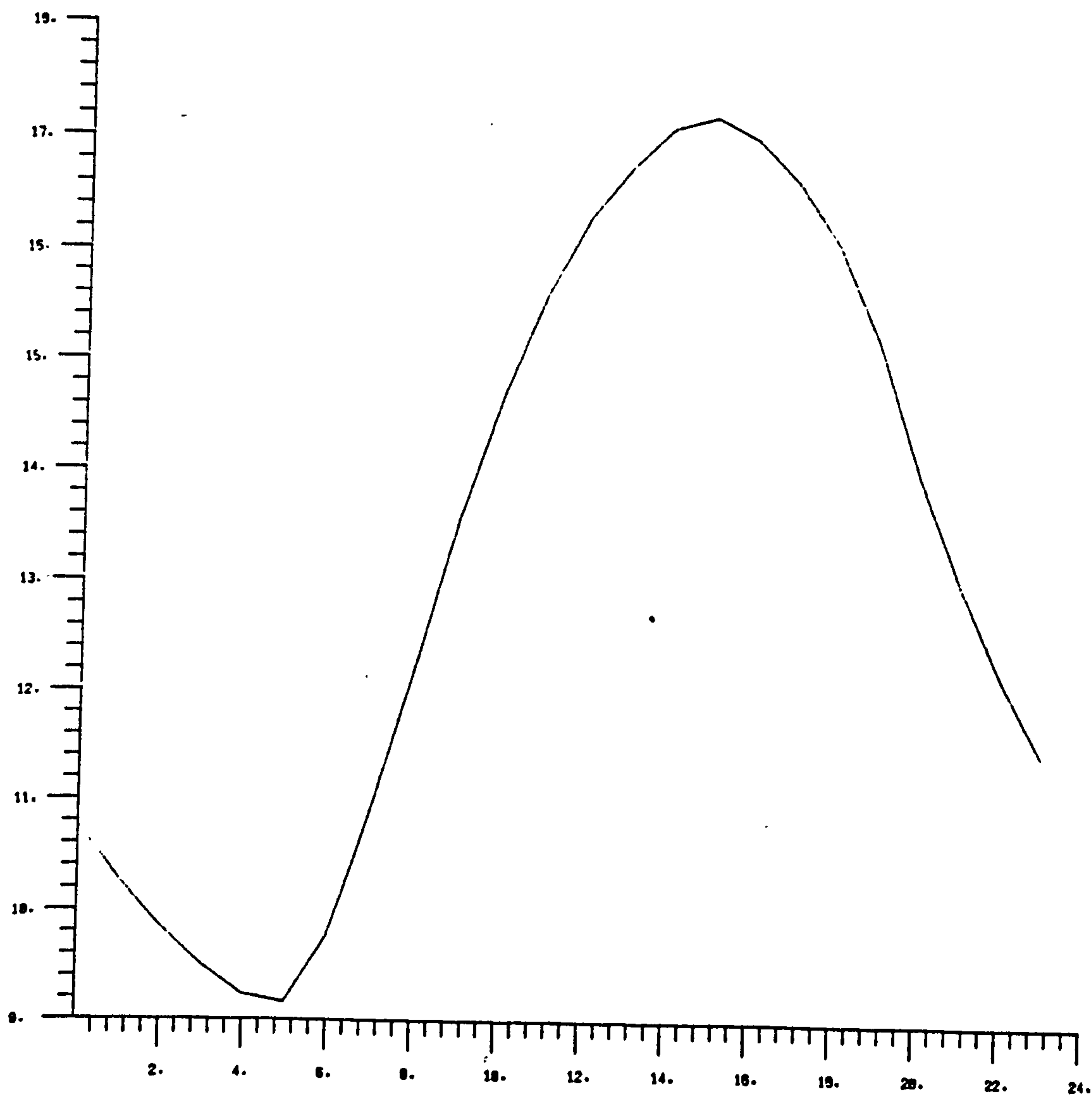
April

GRAPH OF MEAN TEMPERATURES(OVER 25 Y)AGAINST TIME FOR APRIL



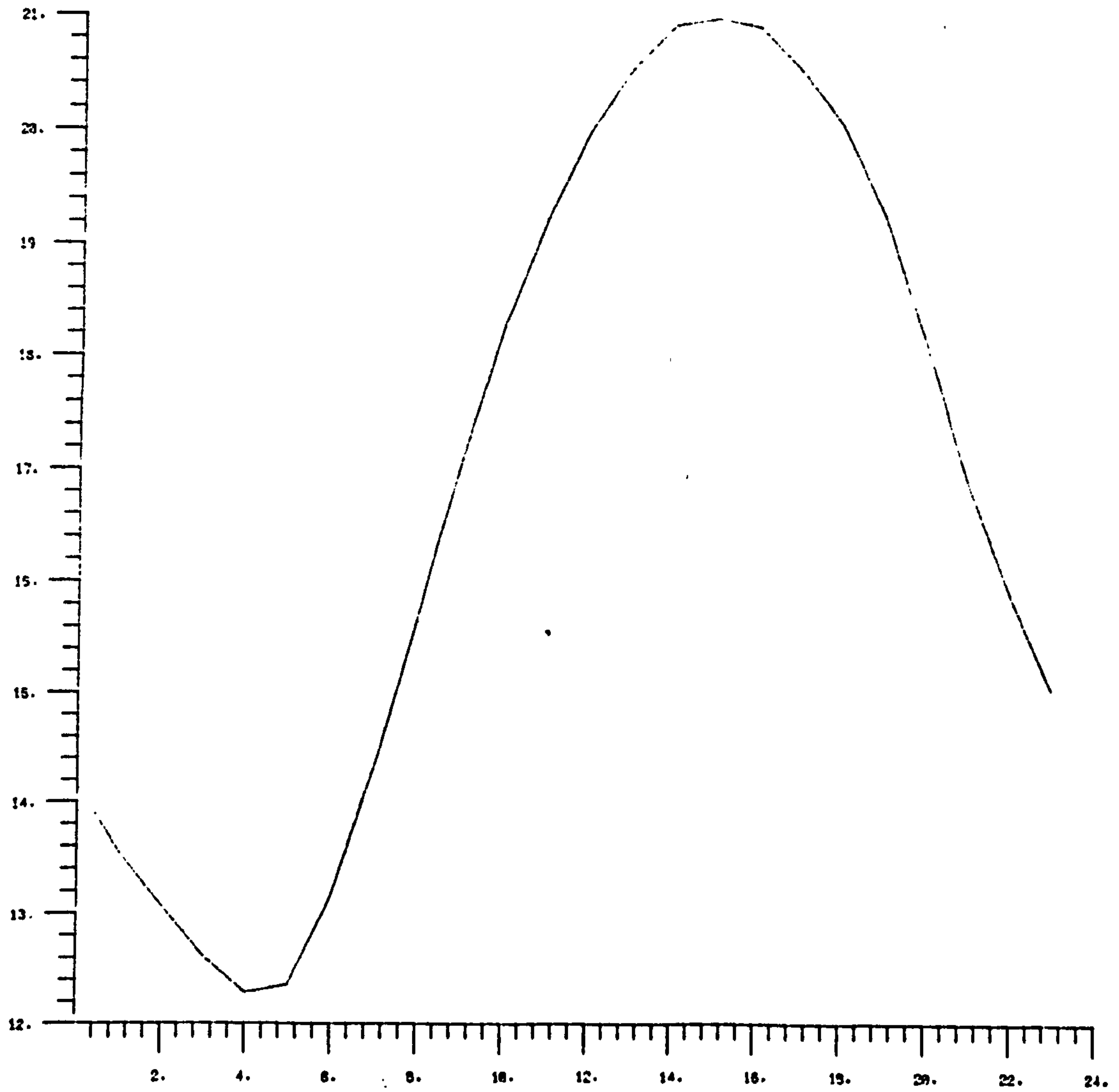
May

GRAPH OF MEAN TEMPERATURES (OVER 25 Y) AGAINST TIME FOR MAY ...



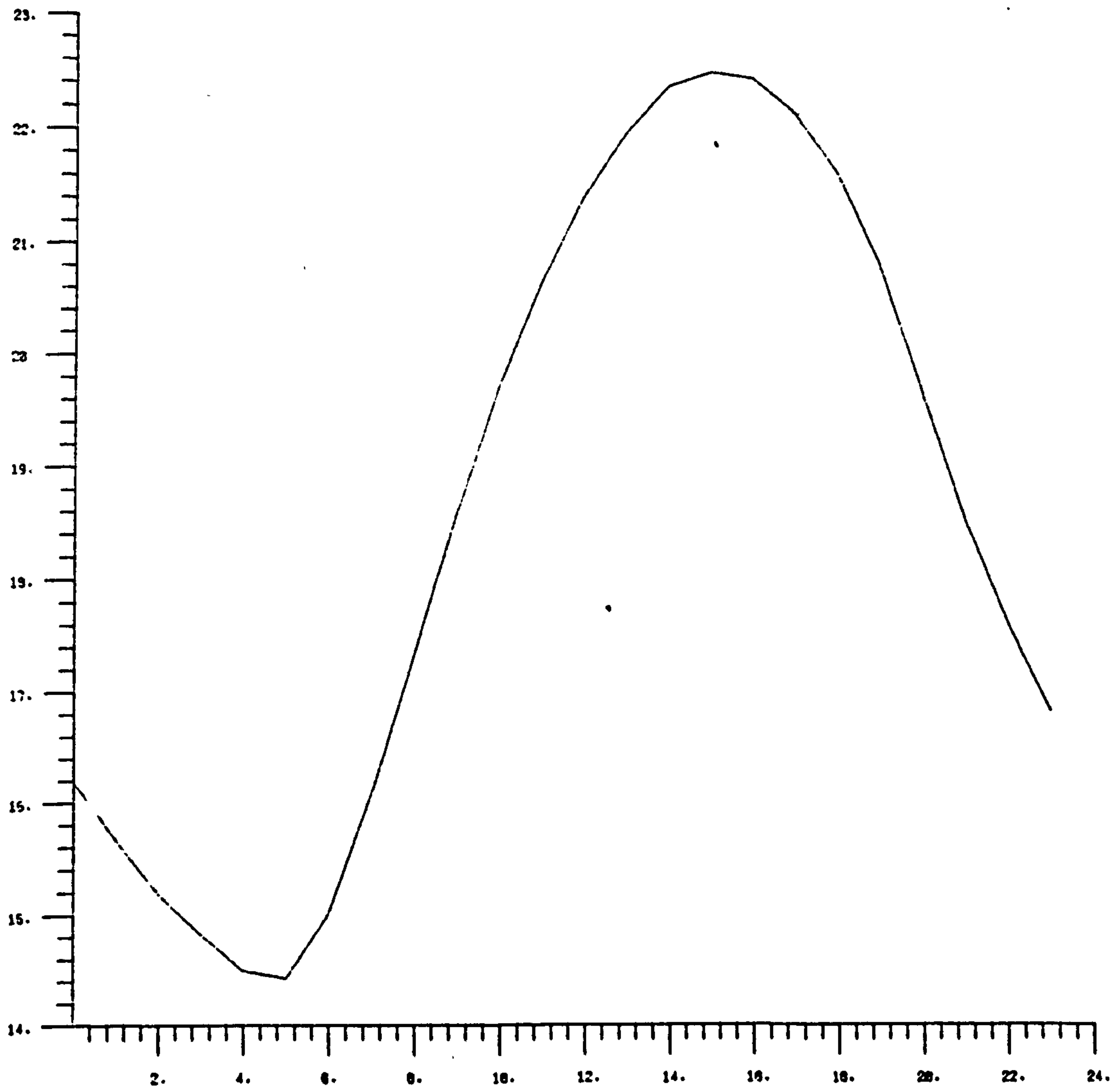
June

GRAPH OF MEAN TEMPERATURES (OVER 25 YEARS) AGAINST TIME FOR JUNE



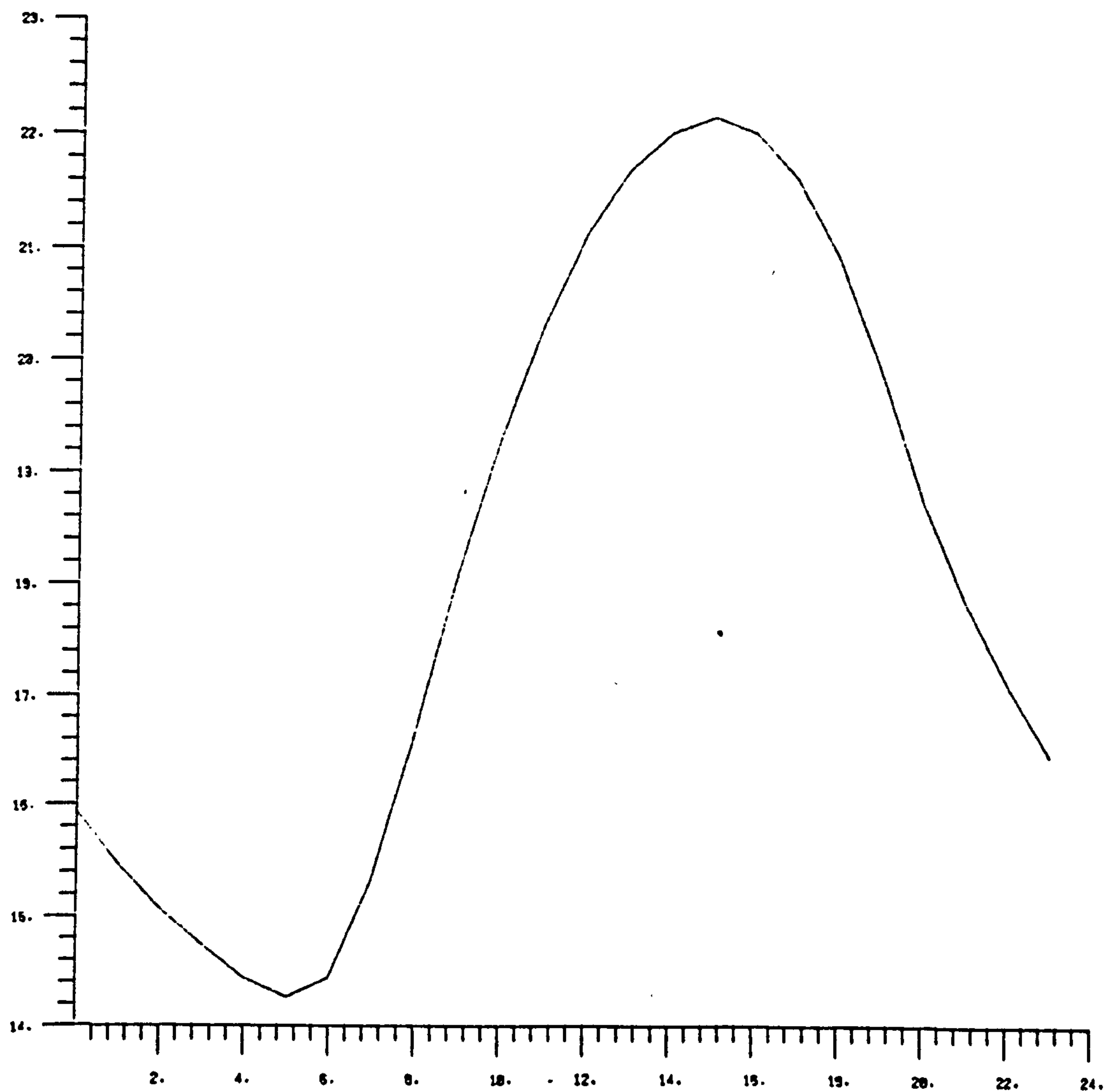
July

GRAPH OF MEAN TEMPERATURES(OVER 25 Y)AGAINST TIME FOR JULY



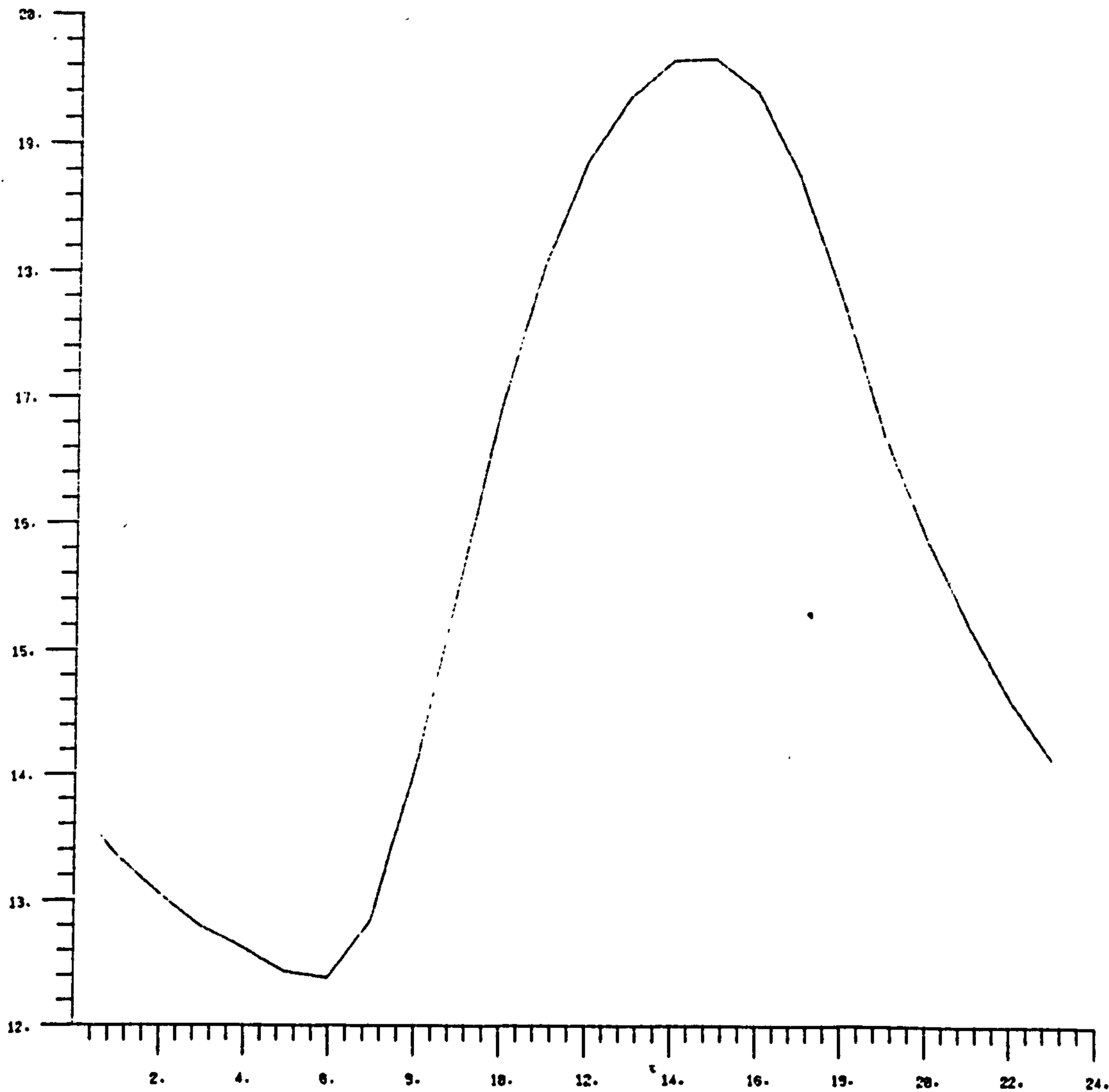
August

GRAPH OF MEAN TEMPERATURES (OVER 25 Y) AGAINST TIME FOR AUGUST



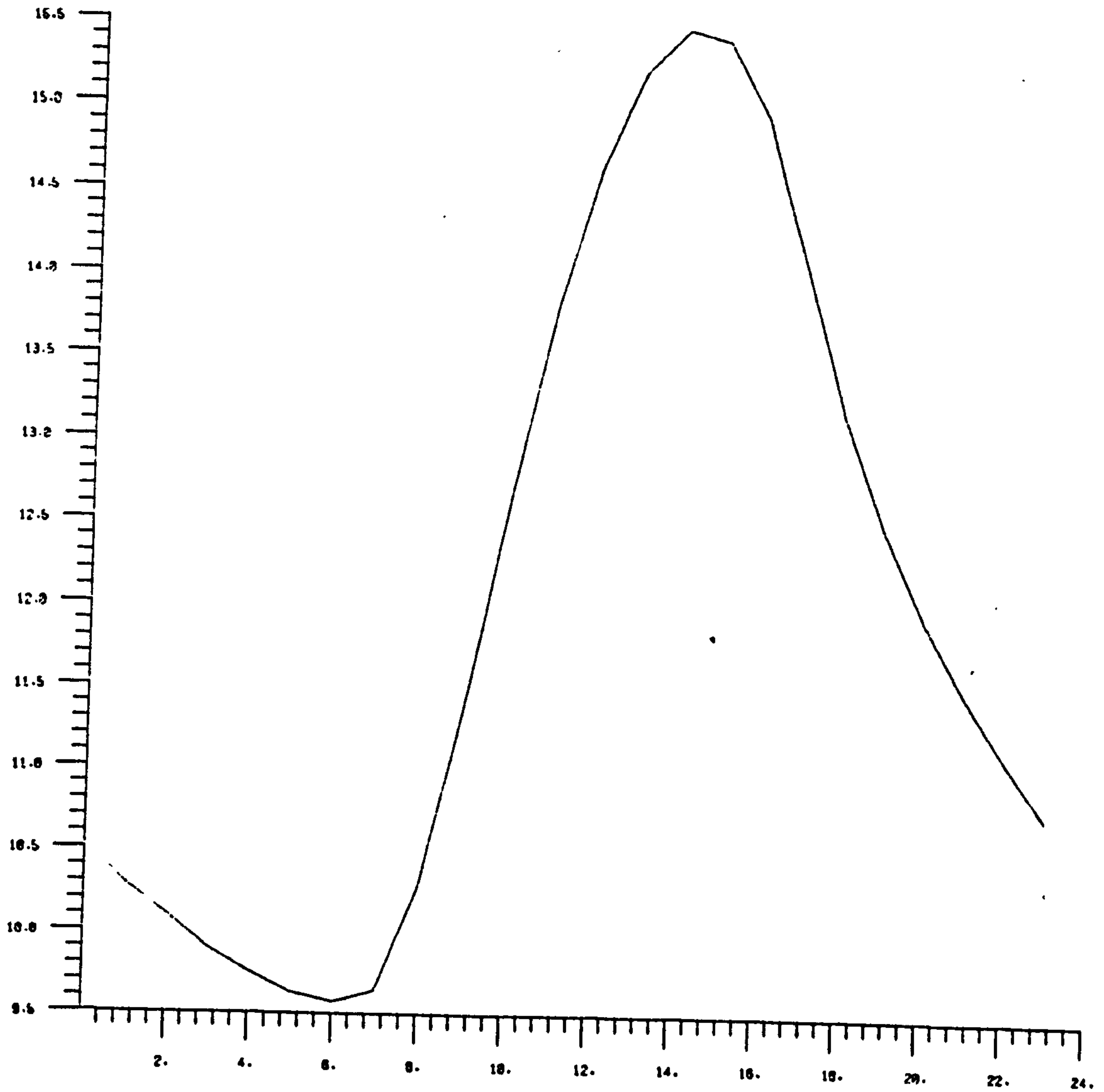
September

GRAPH OF MEAN TEMPERATURES (OVER 25 Y) AGAINST TIME FOR SEPTEMBER



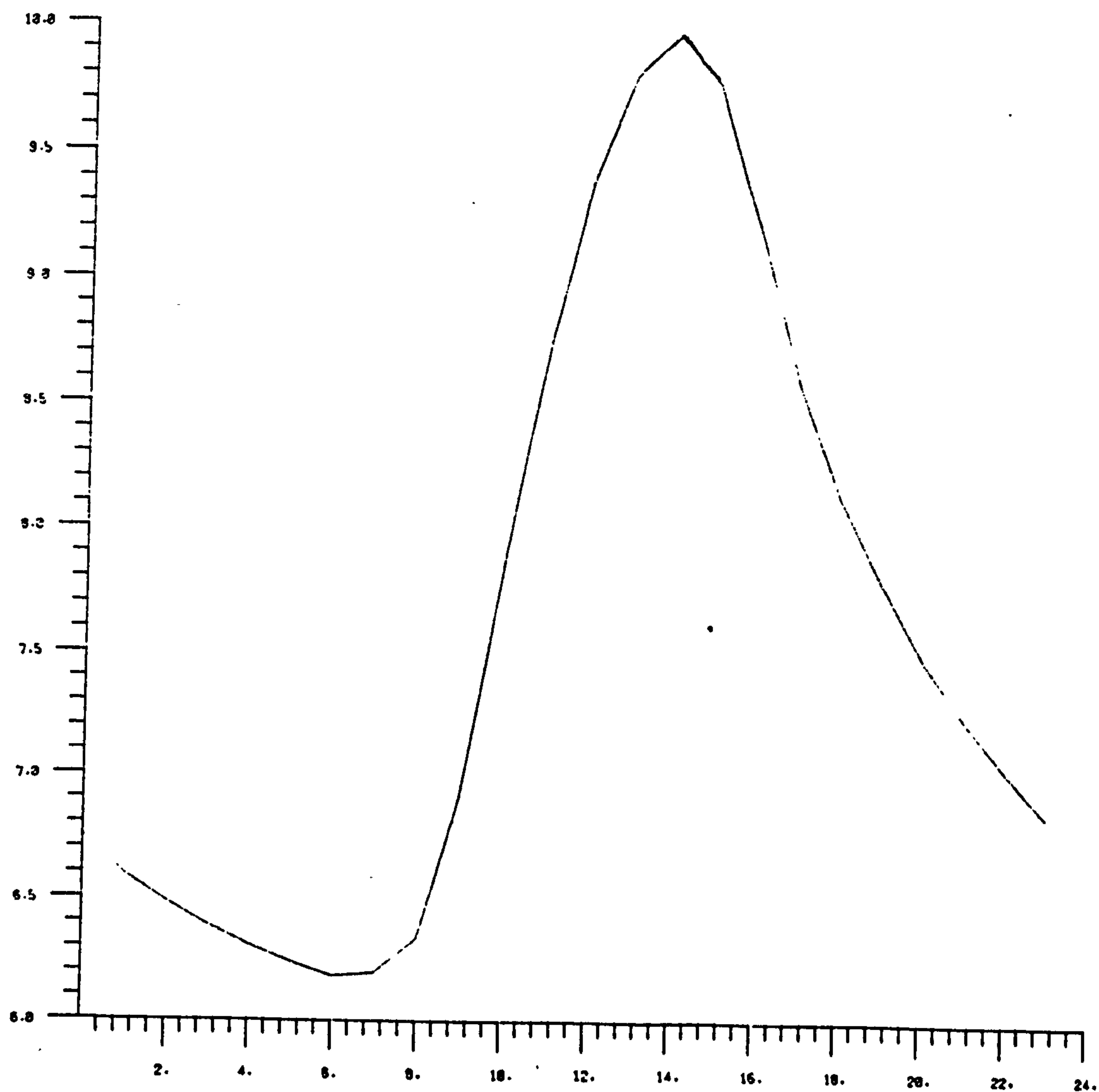
October

GRAPH OF MEAN TEMPERATURES (OVER 25 Y) AGAINST TIME FOR OCTOBER



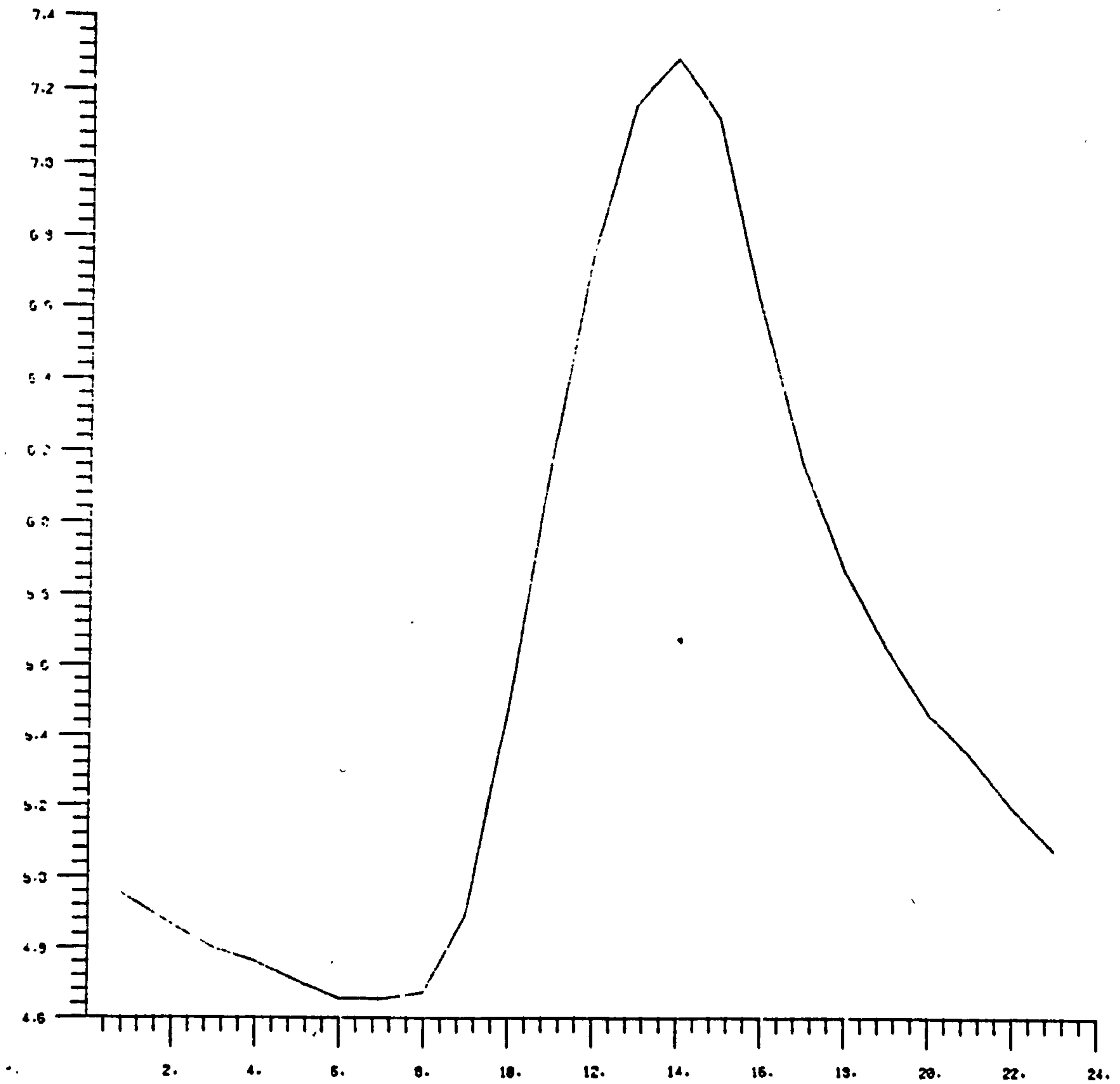
November

GRAPH OF MEAN TEMPERATURES (OVER 25 Y) AGAINST TIME FOR NOVEMBER



December

GRAPH OF MEAN TEMPERATURES (OVER 25 Y) AGAINST TIME FOR DECEMBER



Appendix IV

Computer model of the house/heat pump system using real weather data. Program and example of type of results obtained.

```

1  JOB CM14 3822,SAVE
2  ROUTE OPENUNIV
3  POST CM14
4  TAPE9 CM1402
5  PRINTER 10K
6  COMP 60 SECS
7  PHX
8  //MET DD UNIT=TAPE9,VOL=SER=CM1402,LABEL=(1,SL,,IN),DISP=(OLD,KEEP),
9  //      DSN=CM14.Z,DCB=(RECFM=VS,LRECL=392,BLKSIZE=396,DEN=3)
10 FTG1CLG(FT01F001=MET PROGRAM=XH1 FT08F001=.OUT1/VBA/N FT05F001=XH# ,
11      FT07F001=.OUT2/VBA/N)
12      COMMON/BLOCK1/PRELIM(2),DATA(8,24)
13      COMMON/DD/TI
14      COMMON/BLOCK2/TSET,TAMB,TIME,KSYS,BUFEN,MDHW,HOUT,ELUSE,GASUSE,
15      CTTANK(100),MDOT,MDOT1,MDOT2,DHWCUM,HCUM,TC,TH,TENON,TSTEP,TLAG,
16      DMASS,K,TMIN,HMAX,WPUMPE,FANE,TM,DHWT,MDOT3,CONST,TLOSS,RATE,D,SP,
17      EV,AREA(3),UVAL(3),ITYPE(3),DEN(3),VOL(3),SPHT(3),TUAREA,FLOSS,VENT
18      F,UAREA(3),QHEAT(3),MCP(3),TMCP,TOTMC
19      COMMON/EX/TSTART
20      REAL MDOT,MDOT1,MDOT2,MASS,MDHW,MCP,MCPA,MDOT3
21      INTEGER*2 INT(194),PRELIM,DATA
22      INTEGER T
23      EQUIVALENCE(PRELIM(1),INT(1))
24      TSTEP=0.08333
25      TMCP=0.0
26      K=0
27      TSET=20.0
28      CALL READ
29      TLOSS=6444.0
30      CONST=TLOSS/20.0/1000.0
31      WRITE(8,102)CONST
32      WRITE(7,102)CONST
33      102 FORMAT(6X,'CONSTANT FOR HOUSE =',F10.2)
34      DO 40 I=1,3
35      MCP(I)=DEN(I)*VOL(I)*SPHT(I)
36      TMCP=TMCP+MCP(I)
37      40 CONTINUE
38      MCPA=D*V*SP
39      TOTMC=(TMCP/2.0+MCPA)/(3.6E+6)/TSTEP
40      WRITE(8,104)TOTMC
41      WRITE(7,104)TOTMC
42      104 FORMAT(' TOTMC IS ',F15.8)
43      888 READ(1,END=999)INT
44      TSTART=-100.
45      IF(PRELIM(1).NE.1975)GO TO 888
46      IF(PRELIM(2).NE.1)GO TO 888
47      IF(DATA(1,1).NE.31)GO TO 888
48      WRITE(8,103)PRELIM(1),PRELIM(2),DATA(1,1)
49      WRITE(7,103)PRELIM(1),PRELIM(2),DATA(1,1)
50      103 FORMAT(///,'*****','YEAR IS ',I4,4X,'MONTH ',I2,4X,'DAY ',I2
51      C,'*****')
52      WRITE(8,301)
53      301 FORMAT(T10,'TIME',T20,'TAMB',T30,'KSYS',T40,'BUFEN',T50,'MDOT3
54      C',T60,'HOUT',T70,'TI',T80,'ELUSE',T90,'GASUSE')
55      WRITE(7,303)
56      303 FORMAT(T10,'TTANK(1)',T20,'TTANK(100)',T30,'MDOT',T40,'MDOT1',
57      CT50,'MDOT2',T60,'DHWCUM',T70,'HCUM',T80,'TC',T90,'TH',T100,'TENON
58      C')
59      T=0
60      DO 10 I=1,24
61      TAMB=DATA(5,I)/10.0
62      DO 5 J=1,12
63      777 T=T+5
64      TIME=T/60.
65      IF((TIME.GT.0.0.AND.TIME.LE.6.0)KSYS=0
66      IF((TIME.GT.6.0.AND.TIME.LE.9.0)KSYS=1
67      IF((TIME.GT.9.0.AND.TIME.LE.16.0)KSYS=0
68      IF((TIME.GT.16.0.AND.TIME.LE.24.0)KSYS=1
69      MDHW=0.0
70      IF((TIME.GT.8.0.AND.TIME.LE.8.08333).OR.
71      C(TIME.GT.13.0.AND.TIME.LE.13.08333).OR.
72      C(TIME.GT.19.0.AND.TIME.LE.19.08333).OR.
73      C(TIME.GT.21.0.AND.TIME.LE.21.08333))MDHW=0.094
74      CALL WATER
75      TI=((TI*((2.0*TOTMC)-CONST))+2.0*(HOUT+(CONST*TAMB)))
76      C/((2.0*TOTMC)+CONST)
77      5 CONTINUE
78      CALL WRITE

```

Cont'd


```

79      10 CONTINUE
80      GO TO 888
81      999 WRITE(8,998)
82      998 FORMAT(' END OF TAPE DATA')
83      STOP
84      END
85  C*****
86      SUBROUTINE READ
87      COMMON/DD/TI
88      COMMON/BLOCK2/TSET,TAMB,TIME,KSYS,BUFEN,MDHW,HOUT,ELUSE,GASUSE,
89      CTTANK(100),MDOT,MDOT1,MDOT2,DHWCUM,HCUM,TC,TH,TENON,TSTEP,TLAG,
90      DMASS,K,TMIN,HMAX,WPUMPE,FANE,TM,DHWT,MDOT3,CONST,TLOSS,RATE,D,SP,
91      EV,AREA(3),UVAL(3),ITYPE(3),DEN(3),VOL(3),SPHT(3),TUAREA,FLOSS,VENT
92      F,UAREA(3),QHEAT(3),MCP(3),TMCP,TOTMC
93      REAL MDOT,MDOT1,MDOT2,MASS,MDHW,MDOT3
94      READ(5,201)TAMB,TI
95      201 FORMAT(2F10.0)
96      TGRD=TAMB
97      READ(5,202)RATE,D,SP,V
98      202 FORMAT(4F10.0)
99      DO 20 I=1,3
100     READ(5,203)AREA(I),UVAL(I),ITYPE(I),DEN(I),VOL(I),SPHT(I)
101     203 FORMAT(2F10.0,I10,3F10.0)
102     20 CONTINUE
103     RETURN
104     END
105  C*****
106      SUBROUTINE LOSS
107      COMMON/DD/TI
108      COMMON/BLOCK2/TSET,TAMB,TIME,KSYS,BUFEN,MDHW,HOUT,ELUSE,GASUSE,
109      CTTANK(100),MDOT,MDOT1,MDOT2,DHWCUM,HCUM,TC,TH,TENON,TSTEP,TLAG,
110      DMASS,K,TMIN,HMAX,WPUMPE,FANE,TM,DHWT,MDOT3,CONST,TLOSS,RATE,D,SP,
111      EV,AREA(3),UVAL(3),ITYPE(3),DEN(3),VOL(3),SPHT(3),TUAREA,FLOSS,VENT
112      F,UAREA(3),QHEAT(3),MCP(3),TMCP,TOTMC
113      REAL MDOT,MDOT1,MDOT2,MASS,MDHW,MDOT3,MCP,MCPA
114      DO 3 I=1,3
115     IT=ITYPE(I)
116     UAREA(I)=AREA(I)*UVAL(I)
117     TUAREA=TUAREA+UAREA(I)
118     IF(IT.EQ.1)GO TO 33
119     IF(IT.EQ.2)GO TO 44
120     33 QHEAT(I)=UAREA(I)*(TI-TAMB)
121     GO TO 55
122     44 QHEAT(I)=UAREA(I)*(TI-TGRD)
123     55 FLOSS=FLOSS+QHEAT(I)
124     3 CONTINUE
125     VENT=VENT+(RATE/3600.0)*D*SP*V*(TI-TAMB)
126     TLOSS=TLOSS+VENT
127     RETURN
128     END
129  C*****
130      SUBROUTINE WRITE
131      COMMON/DD/TI
132      COMMON/BLOCK2/TSET,TAMB,TIME,KSYS,BUFEN,MDHW,HOUT,ELUSE,GASUSE,
133      CTTANK(100),MDOT,MDOT1,MDOT2,DHWCUM,HCUM,TC,TH,TENON,TSTEP,TLAG,
134      DMASS,K,TMIN,HMAX,WPUMPE,FANE,TM,DHWT,MDOT3,CONST,TLOSS,RATE,D,SP,
135      EV,AREA(3),UVAL(3),ITYPE(3),DEN(3),VOL(3),SPHT(3),TUAREA,FLOSS,VENT
136      F,UAREA(3),QHEAT(3),MCP(3),TMCP,TOTMC
137      REAL MDOT,MDOT1,MDOT2,MASS,MDHW,MDOT3,MCP,MCPA
138      WRITE(8,302)TIME,TAMB,KSYS,BUFEN,MDOT3,HOUT,TI,ELUSE,GASUSE
139      302 FORMAT(T10,F6.2,T20,F6.2,T32,I1,T40,F6.2,T50,F6.3,T58,F6.2,T70,
140      CF6.2,T78,F6.2,T88,F8.2)
141      WRITE(7,304)TTANK(1),TTANK(100),MDOT,MDOT1,MDOT2,DHWCUM,HCUM,TC,
142      CTH,TENON
143      304 FORMAT(T10,F6.2,T20,F6.2,T28,F6.2,T38,F6.2,T48,F6.2,T58,F8.2,T68,
144      CF8.2,T78,F6.2,T88,F6.2,T98,F6.2)
145      RETURN
146      END
147  C*****
148      SUBROUTINE WATER
149      COMMON/DD/TROOM
150  C DUMMY VARIABLES (IN ORDER)
151  C -----
152  C TROOM = ROOM TEMP. (DEG.C)
153  C TSET = THERMOSTAT SETTING (DEG.C)
154  C TAMB = OUTSIDE DRY BULB TEMPERATURE (DEG.C)
155  C TIME = TIME OF DAY (DECIMAL HOURS)
156  C KSYS = 0 WHEN TIME CLOCK IS 'OFF'
157  C       1 WHEN TIME CLOCK IS 'ON'.
158  C COMMON BLOCK (IN ORDER)
159  C -----
160  C TSTEP = TIME STEP (DECIMAL HOURS)

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161 C TLAG = THERMOSTAT LAG (ON TO OFF) (DEG.C)
162 C BUFEN = ENTHALPY OF BUFFER WATER ABOVE 35 DEG.C (KWH)
163 C ELUSE = CUMULATIVE ELECTRICITY USE (KWH)
164 C GASUSE = CUMULATIVE GAS USE (KWH)
165 C HOUT = POWER TO HOUSE IN LAST TIME STEP(KW)
166 C TTANK(100) = BUFFER TEMP. FROM TOP(1) TO BOTTOM(100) (DEG.C)
167 C MDOT = WATER FLOW RATE (TOTAL) (KG/SEC)
168 C MDOT1 = WATER FLOW RATE (TO RADIATORS) (KG/SEC)
169 C MDOT2 = WATER FLOW RATE ( TO BUFFER) (KG/SEC)
170 C MDOT3 =WATER FLOW RATE( TO DHW)
171 C MASS = BUFFER STORE MASS (KG)
172 C K = FLAG, INITIALISED AS ZERO IN MASTER SEGMENT.
173 C HMAX = HEAT LOSS OF HOUSE FOR TROOM=20, TAMB=0 (KW)
174 C WPUMPE = WATER PUMP RATING (W)
175 C HCUM = CUMULATIVE HEAT TO HOUSE (KWH)
176 C TC =WATER ENTRY TEMP. TO CONDENSER (DEG.C)
177 C TM =WATER EXIT TEMP. TO CONDENSER (DEG.C)
178 C TH = WATER EXIT TEMP. FROM EXHAUST HEAT EXCHANGER (DEG.C)
179 C DHWCUM = CUMULATIVE DHW USE(KWH)
180 C DHWT = DHW TEMP (DEG.C)
181 C TENON = CUMULATIVE TIME ENGINE IS ON (HOURS)
182 COMMON/BLOCK2/TSET,TAMB,TIME,KSYS,BUFEN,MDHW,HOUT,ELUSE,GASUSE,
183 CTTANK(100),MDOT,MDOT1,MDOT2,DHWCUM,HCUM,TC,TH,TENON,TSTEP,TLAG,
184 DMASS,K,TMIN,HMAX,WPUMPE,FANE,TM,DHWT,MDOT3,CONST,TLOSS,RATE,D,SP,
185 EV,AREA(3),UVAL(3),ITYPE(3),DEN(3),VOL(3),SPHT(3),TUAREA,FLOSS,VENT
186 F,UAREA(3),QHEAT(3),MCP(3),TMCP,TOTMC
187 COMMON/EX/TSTART
188 REAL MDOT,MDOT1,MDOT2,MASS,MDHW,MDOT3
189 IF(K.GT.0) GO TO 1
190 K=1
191 TMIN=35.0
192 DHWT=0.
193 TH=0.0
194 TM=46.5
195 TLAG=1
196 TENON=0.
197 BUFEN=0.
198 ELUSE=0.
199 GASUSE=0.
200 HMAX=7.0
201 WPUMPE=100.
202 DHWCUM=0.
203 HCUM=0.
204 MASS=2000.
205 DO 2 I=1,100
206 2 TTANK(I)=TMIN
207 1 IF(K.EQ.1) TONOFF=TSET-TLAG/2.
208 IF(K.EQ.2) TONOFF=TSET+TLAG/2.
209 C TONOFF IS THE TEMP. AT WHICH THE HEATING SWITCHES ON OR OFF. IT DEPENDS
210 C ON THE THERMOSTAT LAG AND THE SET TEMPERATURE.
211 K=1
212 IF(TROOM.LT.TONOFF .AND. KSYS.EQ.1) K=2
213 C IF K=1, HEATING IS OFF , IF K=2 HEATING IS ON.
214 C FIND IF HEAT PUMP IS ON AND SEND TO "PUMP ON" ROUTINE.
215 IF( (TIME.GE.13.0 .AND. TIME.LE.17.0 .AND. TTANK(100).LT.55.0)
216 1.OR. (TTANK(1).LE.45. .AND. (K.EQ.2 .OR. (KSYS.EQ.1 .AND.
217 2 MDHW.GT.0.0))) GO TO 3
218 IF((TIME-TSTART).LE.0.5.AND.(TIME.LT.13.0.OR.TIME.GT.17.0))
219 CGO TO 31
220 TSTART=-100.0
221 C ROUTINE FOR PUMP OFF
222 C-----
223 T=0.
224 MDOT=0.
225 MDOT1=0.
226 MDOT2=0.
227 MDOT3=0.
228 HOUT=0.
229 IF(K.EQ.1) GO TO 4
230 C HEATING ON
231 8 TSUPPLY=TTANK(1)
232 TRET=TMIN
233 TDIF=((TSUPPLY+TRET)/2.0)-TROOM
234 POWER=HMAX*((1.8*TDIF)-9.0)/35.
235 IF(TSUPPLY.EQ.TRET)GO TO 32
236 MDOT1=POWER/(4.1868*(TSUPPLY-TRET))
237 IF(MDOT1.LE.0.1) GO TO 5
238 32 MDOT1=0.1
239 TKET=HMAX*1.8/35./4.1868/0.1
240 TKET=(TSUPPLY*(1.-TRET/2.))+TRET*(TROOM+5.)/(1.+TRET/2.)
241 POWER=0.1*4.1868*(TSUPPLY-TRET)
242 C SUPPLY OF DHW ALSO
243 C-----
244 IF(MDHW.EQ.0.)GO TO 5
245 MDOT3=MDHW
246 MDOT=MDOT1+MDOT3
247 TRET=(MDOT1*TRET+MDOT3*15.)/MDOT
248 GO TO 105

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249      5 MDOT=MDOT1
250      105 DT=MASS/MDOT/360000.
251      C TIME IN HOURS TO SHIFT THE SLUG.
252          T=T+DT
253          IF(T.GT.TSTEP) DT=DT-T+TSTEP
254          HOUT=HOUT+POWER*DT
255          BUFEN=BUFEN-(TSUPPLY-TRET)*4.1868*MDOT*DT
256          DHWCUM=DHWCUM+4.1868*MDOT3*(TSUPPLY-15.)*DT
257          IF(T.GT.TSTEP) GO TO 6
258          DO 7 I=1,99
259      7   TTANK(I)=TTANK(I+1)
260          TTANK(100)=TRET
261          GO TO 8
262      6   X=DT/MASS*MDOT*360000.
263      C X IS FRACTION OF SLUG MOVED
264          DO 9 I=1,99
265      9   TTANK(I)=(1.-X)*TTANK(I) + X*TTANK(I+1)
266          TTANK(100)=(1.-X)*TTANK(100)+X*TRET
267          ELUSE=ELUSE+WPUMPE/1000.*TSTEP
268          HCUM=HCUM+HOUT
269          HOUT=HOUT/TSTEP
270          DHWT=TSUPPLY-5.
271          RETURN
272      C   SUPPLY OF DHW ONLY
273      C -----
274      4   IF(MDHW.EQ.0.) RETURN
275          ELUSE=ELUSE+WPUMPE/1000.*TSTEP
276          MDOT3=MDHW
277          MDOT=MDOT3
278      10  TSUPPLY=TTANK(1)
279          TRET=15.0
280
281          DT=MASS/MDOT/360000.
282          T=T+DT
283          IF(T.GT.TSTEP) DT=DT-T+TSTEP
284          DHWCUM=DHWCUM+MDOT*4.1868*(TSUPPLY-TRET)*DT
285          BUFEN=BUFEN-MDOT*4.1868*(TSUPPLY-TRET)*DT
286          IF(T.GT.TSTEP) GO TO 11
287          DO 12 I=1,99
288      12  TTANK(I)=TTANK(I+1)
289          TTANK(100)=TRET
290          GO TO 10
291      11  X=DT/MASS*MDOT2*360000.0
292          DO 13 I=1,99
293      13  TTANK(I)=(1.-X)*TTANK(I) + X*TTANK(I+1)
294          TTANK(100)=(1.-X)*TTANK(100)+X*TRET
295          DHWT=TSUPPLY-5.
296          RETURN
297      C HEAT PUMP ON ROUTINE
298      C -----
299      3   TSTART=TIME
300      31  TENON=TENON+TSTEP
301          T=0.
302          HOUT=0.
303          MDOT1=0.
304          RPM=830.
305          TEV=TAMB-8.0
306          C=RPM/1500.*(TEV*TEV*0.00585+TEV*0.4985+12.48)
307          P=RPM/1500.*(0.0733*TEV+3.95)
308          ETA=(4.5*P/0.7457+9.1)/100.
309          E=6./7.*P*(1./ETA-1.)
310          GASUSE=GASUSE+P/ETA*TSTEP
311          ELUSE=ELUSE+WPUMPE/1000.*TSTEP
312          IF(K.EQ.1) GO TO 16
313      C HEATING ON ROUTINE
314      18  IF(TTANK(100).EQ.35.0)GO TO 14
315          S=HMAX*(35.-TTANK(100.))/35./4.1868
316          TT=TM-TTANK(100)
317          U=C/4.1868
318          V=TM-35.
319          W=E/4.1868
320          X=0.9*TM-1.8*TROOM+22.5
321          A=TT*V
322          B=W*TT-U*V-S*X
323          CC=-U*W-0.9*S*W
324          BBC=B*B-4.*A*CC
325          IF((B*B-4.*A*CC).LE.0.0)GO TO 55
326          MDOT=(-B+SQRT(B*B-4.*A*CC))/2./A
327          GO TO 15
328      55  HOUT=HOUT/TSTEP
329          MDOT=0.0
330          MDOT1=0.0
331          MDOT2=0.0
332          MDOT3=0.0
333          CALL WRITE
334          RETURN

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335      14 MDOT=C/4.1868/(TM-35.)
336      15 TC=TM-C/4.1868/MDOT
337      TH=TM+E/4.1868/MDOT
338      MDOT1=HMAX*(0.9*TH-1.8*TR00M+22.5)/35./4.1868/(TH-35.)
339      MDOT3=MDHW
340      IF(MDOT1.GE.MDOT)MDOT1=MDOT-0.00001.
341      MDOT2=MDOT-MDOT1-MDOT3
342      DT=MASS/MDOT2/360000.0
343      T=T+DT
344      IF(T.GT.TSTEP) DT=DT-T+TSTEP
345      HOUT=HOUT+MDOT1*4.1868*(TH-35.)*DT
346      HCUM=HCUM+MDOT1*4.1868*(TH-35.)*DT
347      DHWCUM=DHWCUM+MDOT3*(TH-15.)*4.1868*DT
348      DHWT=TH-5.
349      BUFEN=BUFEN+MDOT2*4.1868*DT*(TH-TTANK(100))
350      IF(T.GE.TSTEP) GO TO 19
351      DO 17 I=1,99
352      J=101-I
353      17 TTANK(J)=TTANK(J-1)
354      TTANK(1)=TH
355      IF(TTANK(100).GE.55.) HOUT=HOUT/TSTEP
356      IF(TTANK(100).GE.55.) RETURN
357      GO TO 18
358      19 HOUT=HOUT/TSTEP
359      X=DT/MASS*MDOT2*360000.0
360      DO 20 I=1,99
361      J=101-I
362      20 TTANK(J)=(1.-X)*TTANK(J)+X*TTANK(J-1)
363      TTANK(1)=(1.-X)*TTANK(1)+X*TH
364      RETURN
365 C   BUFFER CHARGE, HEATING OFF
366 C   -----
367      16 IF(MDHW.GT.0.0)GO TO 511
368      MDOT1=0.
369      MDOT3=0.
370      HOUT=0.
371      T=0.
372      23 MDOT=C/4.1868/(TM-TTANK(100))
373      IF(MDOT.GT.0.0)GO TO 66
374      MDOT2=0.0
375      MDOT=0.0
376      CALL WRITE
377      RETURN
378      66 TH=TM+E/MDOT/4.1868
379      MDOT2=MDOT
380      DT=MASS/MDOT2/360000.0
381      T=T+DT
382      IF(T.GT.TSTEP) DT=DT-T+TSTEP
383      BUFEN=BUFEN+MDOT2*4.1868*DT*(TH-TTANK(100))
384      IF(T.GE.TSTEP) GO TO 21
385      DO 22 I=1,99
386      J=101-I
387      22 TTANK(J)=TTANK(J-1)
388      TTANK(1)=TH
389      IF(TTANK(100).GE.55.) RETURN
390      GO TO 23
391      21 X=DT/MASS*MDOT2*360000.0
392      DO 24 I=1,99
393      J=101-I
394      24 TTANK(J)=(1.-X)*TTANK(J)+X*TTANK(J-1)
395      TTANK(1)=(1.-X)*TTANK(1)+X*TH
396      RETURN
397 C   SUPPLY OF DHW
398      511 MDOT1=0.
399      MDOT3=MDHW
400      123 MDOT=(C/4.1868-MDHW*(TTANK(100)-15.))/(TM-TTANK(100))
401      MDOT2=MDOT-MDOT3
402      DT=MASS/MDOT2/360000.
403      T=T+DT
404      IF(T.GT.TSTEP)DT=DT-T+TSTEP
405      DHWT=TH-5.
406      DHWCUM=DHWCUM+4.1868*MDHW*(TH-15.)*DT
407      BUFEN=BUFEN+MDOT2*4.1868*DT*(TH-TTANK(100))
408      IF(T.GE.TSTEP)GO TO 121
409      DO 122 I=1,99
410      J=101-I
411      122 TTANK(J)=TTANK(J-1)
412      TTANK(1)=TH
413      IF(TTANK(100).GE.55.)RETURN
414      GO TO 123
415      121 X=DT/MASS/MDOT2/360000.
416      DO 124 I=1,99
417      J=101-I
418      124 TTANK(J)=(1.-X)*TTANK(J)+X*TTANK(J-1)
419      TTANK(1)=(1.-X)*TTANK(1)+X*TH
420      RETURN
421      END
422

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Cont'd

423	0.0	20.0					
424	1.0	1.293	993.0	187.0			
425	126.4	0.996	1	1250.0	34.76	900.0	
426	11.22	5.6	1	0.0	0.00	0.0	
427	37.76	0.818	2	0.0	0.00	0.0	
428	*						

Results

An example of the type of output produced by the simulation is given below. An explanation of the parameters used is given in the main text on page 98.

YEAR IS 1975	MONTH 1	DAY 31***						
TIME	TAMB	KSYS	BUFEN	MDOT3	HOUT	TI	ELUSE	GASUSE
1.00	11.40	0	0.0	0.0	0.0	19.51	0.0	0.0
2.00	11.20	0	0.0	0.0	0.0	19.04	0.0	0.0
3.00	10.70	0	0.0	0.0	0.0	18.56	0.0	0.0
4.00	10.50	0	0.0	0.0	0.0	18.10	0.0	0.0
5.00	10.90	0	0.0	0.0	0.0	17.69	0.0	0.0
6.00	10.90	0	0.0	0.0	0.0	17.31	0.0	0.0
7.00	11.00	1	0.25	0.0	7.08	18.38	0.10	5.85
8.00	11.30	1	0.87	0.0	7.48	19.36	0.20	11.70
9.00	11.00	1	1.83	0.0	7.38	20.20	0.30	17.55
10.00	10.80	0	1.83	0.0	0.0	19.66	0.30	17.55
11.00	9.90	0	1.83	0.0	0.0	19.11	0.30	17.55
12.00	9.60	0	1.83	0.0	0.0	18.57	0.30	17.55
13.00	9.40	0	2.98	0.0	0.0	18.04	0.31	18.37
14.00	9.30	0	16.82	0.0	0.0	17.55	0.41	28.28
15.00	9.10	0	30.60	0.0	0.0	17.07	0.51	38.16
16.00	8.80	0	44.28	0.0	0.0	16.60	0.61	48.03
17.00	8.40	1	49.23	0.0	8.28	17.65	0.71	57.86
17.17	8.00	1	50.14	0.0	7.66	17.72	0.72	59.49
17.25	8.00	1	50.14	0.0	0.0	17.79	0.73	60.31
17.33	8.00	1	50.14	0.0	0.0	17.75	0.74	61.13
17.42	8.00	1	50.14	0.0	0.0	17.70	0.75	61.95
17.50	8.00	1	50.14	0.0	0.0	17.65	0.76	62.76
18.00	8.00	1	46.08	0.0	8.31	18.05	0.81	62.76
19.00	6.90	1	37.94	0.0	8.01	18.86	0.91	62.76
20.00	6.80	1	30.07	0.0	7.74	19.56	1.01	62.76
21.00	5.00	1	22.45	0.0	7.54	20.08	1.11	62.76
22.00	4.60	1	15.00	0.0	7.37	20.51	1.21	62.76
23.00	4.00	1	15.00	0.0	0.0	19.57	1.21	62.76
24.00	3.50	1	7.98	0.0	7.59	19.90	1.30	62.76

YEAR IS 1975	MONTH 1	DAY 31***							
TTANK(1)	TTANK(100)	MDOT	MDOT1	MDOT2	DHWCUM	HCUM	TC	TH	TENON
35.00	35.00	0.0	0.0	0.0	0.0	0.0	-0.08	0.0	0.0
35.00	35.00	0.0	0.0	0.0	0.0	0.0	-0.08	0.0	0.0
35.00	35.00	0.0	0.0	0.0	0.0	0.0	-0.08	0.0	0.0
35.00	35.00	0.0	0.0	0.0	0.0	0.0	-0.08	0.0	0.0
35.00	35.00	0.0	0.0	0.0	0.0	0.0	-0.08	0.0	0.0
35.00	35.00	0.0	0.0	0.0	0.0	0.0	-0.08	0.0	0.0
43.70	34.01	0.10	0.10	0.0	0.0	8.13	35.00	56.30	0.58
51.75	35.00	0.09	0.09	0.0	0.0	15.95	35.00	56.20	1.17
56.08	35.00	0.08	0.08	0.0	0.0	23.38	35.00	56.30	1.75
56.08	35.00	0.0	0.0	0.0	0.0	23.38	35.00	56.30	1.75
56.08	35.00	0.0	0.0	0.0	0.0	23.38	35.00	56.30	1.75
56.08	35.00	0.0	0.0	0.0	0.0	23.38	35.00	56.30	1.75
56.85	35.00	0.15	0.0	0.15	0.0	23.38	35.00	56.85	1.83
56.89	35.00	0.15	0.0	0.15	0.0	23.38	35.00	56.89	2.83
56.96	35.00	0.15	0.0	0.15	0.0	23.38	35.00	56.96	3.83
57.07	35.00	0.15	0.0	0.15	0.0	23.38	35.00	57.07	4.83
56.15	39.24	0.17	0.10	0.08	0.0	31.99	36.90	55.45	5.83
51.99	46.60	0.0	0.0	0.0	0.0	33.31	40.70	51.99	6.00
51.99	46.60	0.0	0.0	0.0	0.0	33.31	40.70	51.99	6.08
51.99	46.60	0.0	0.0	0.0	0.0	33.31	40.70	51.99	6.17
51.99	46.60	0.0	0.0	0.0	0.0	33.31	40.70	51.99	6.25
57.17	35.00	0.09	0.09	0.0	0.0	37.37	40.70	51.99	6.33
57.07	35.00	0.09	0.09	0.0	0.0	45.51	40.70	51.99	6.33
57.02	35.00	0.08	0.08	0.0	0.0	53.37	40.70	51.99	6.33
56.96	35.00	0.08	0.08	0.0	0.0	61.00	40.70	51.99	6.33
56.91	35.00	0.08	0.08	0.0	0.0	68.45	40.70	51.99	6.33
56.91	35.00	0.0	0.0	0.0	0.0	68.45	40.70	51.99	6.33
56.88	35.00	0.08	0.08	0.0	0.0	75.46	40.70	51.99	6.33

Appendix V

Costs of major components . (1977 prices)

	Type	Supplied by	Cost £
Engine	5 H.P. single cylinder 4 stroke, water cooled	Brit. Engineering	439.60
Compressor	Hallmark model 3c plus flywheel and stop valve	Hall Thermotank Products Ltd.	475.00
Evaporator		Searle Manufacturing Company Ltd.	241.00
Condenser	Bitzer type VW, water cooled	Refrigeration Spares Ltd.	167.15
Oil Separator	GG 108 G (1.7 ton)	Refrigeration Spares Ltd.	25.00
Expansion Valve	Alco TCE 4FW	Refrigeration Spares Ltd.	10.28
Pressure Gauges	R 12 30-120psi (suction) P12 0-400 psi (delivery)	Refrigeration Spares Ltd.	5.05 5.05
Evaporator fan		Airscrew Holden Ltd.	217.00
Crankcase pressure regulator	Danfoss CPL 22		30.31
Differential thermostat	Danfoss RT 270		65.45
Low pressure control	Danfoss RT1		37.85
Hot gas bypass valve	Danfoss EVJD20		37.85
Engine governor	Heinzmann	Industrial Power Units	53.50
Sight glass	SH1, SH2	Refrigeration Spares Ltd.	4.00
Filter drier	Alco EK 053	Refrigeration Spares Ltd.	2.73
Starter motor	Lucas (solenoid)		78.00
Battery	12v AVM 15T		75.00
Carburettor	Impco		25.00
Gas control valve			2.50
Refrigerant 12	Freon 5 lbs	I.C.I.	5.00